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THE ELECTRIC LIGHTING PLANT OF THE U. S. S. *TRENTON*.

FIRST U. S. NAVY INSTALLATION.

By LIEUTENANT-COMMANDER A. M. CHARLTON,
U. S. N., MEMBER.

NOTE: The information in this article was compiled from reports from the *Trenton*, contemporary literature and recollections of officers who served on the vessel. The meagerness of Navy reports is indicated by the following extracts from the ship's log on the first day of commission after the electric plant was installed:

September 18, 1883,—

4-8 P. M.—Testing electric lights.

M. L. WOOD.

8 P. M. to Mid. At 10 P. M. stopped dynamo and banked fire in donkey boiler.

W. T. SWINBURNE.

The present day electrification of machinery on naval vessels when practically every moving element receives its motion from an electric motor, and the vessel throughout is illuminated with electric lamps, makes it very difficult to visualize conditions less than forty years ago when there were no electric lights, electric fans or electric auxiliaries. To strip the *New Mexico,* for instance, of her electric plant would be to take away her motive power, her ability to direct, serve and fire her guns, her means of steering and getting up anchor, and the multitudinous systems of interior and exterior communication.

True enough, our vessels forty years ago moved and fired their guns and were lighted, but their arrangements would seem of the crudest sort to-day.

The installation of an incandescent lighting plant on the U. S. S. *Trenton* in 1883 was the first step in the rapid electrical development which has culminated to-day in our electric drive ships. Before this time there were small interior communication systems such as call bells, thermostats and gun and torpedo-firing systems, but the power involved was negligible and was supplied by wet-cell primary batteries or small hand-driven generators (the Farmer machine on short circuit gave an output of 48 watts, 1/15 H.P.). An engine-driven generator of 5 or 6 horsepower, giving light for an arc lamp, was tried on the *Hartford* in 1877-78 for signalling purposes, but it never got beyond the experimental stage and the successful development of the Very signal system put it in the discard. It has been forgotten even by the people who served on the *Hartford* during its trial.

The fact that the *Trenton* was the first American man-of-war to be fitted with incandescent lamps, and that the installation was made to ascertain the practical efficiency of an electric plant on naval vessels, naturally excited considerable comment, both in and out of the service. One or two passenger vessels and the U. S. Fish Commission Steamer *Albatross* had been fitted with electric plants before the *Trenton* was so equipped, but the latter was a man-of-war, and there was considerable

head-shaking. Fear of fires and of failure to withstand the shock of gun-fire, and the usual sea-going opposition to innovations had to be overcome before the work was authorized.

The installation on the *Albatross* was supervised by Passed Assistant Engineer G. W. Baird, U. S. N. (now Rear Admiral, U. S. N., Retired). The contract was made by him and let without competition. There were 154 lamps of 8 candle-power on this vessel.

When it was decided to put an electric plant on the *Trenton*, Lieut.-Comdr. Royal B. Bradford, U. S. N. (late Rear Admiral), who had had much to do with electricity in connection with torpedoes, was put in charge of the work. Passed Assistant Engineer Baird gave him a copy of the contract for the plant on the *Albatross*, and it was Comdr. Bradford's desire to put in a plant on the *Trenton* double the size of that on the *Albatross* as nearly as possible. Early in 1883 seven electric light companies were asked to bid on the installation, but only the Edison Company had the courage or desire to attempt the work as called for by the Bureau of Navigation circular.

The contract was awarded to the Edison Electric Lighting Company at a cost of $5,500. The contract called for one dynamo and engine complete to supply light to the following lamps: 104 16-candle-power lamps, 130 10-candle-power lamps, and 4 32-candle-power lamps, a total of 238 lamps and 3,092 candle power; 238 key sockets for lamps, 6 extra brushes, 1 automatic regulator, and 1 dynamo foundation. All of the necessary conductors, switches, safety devices and appliances were to be of the most approved patterns then used by the Edison Company; and the fixtures for the lights were to be furnished by the Department. This seems a very loosely drawn contract to-day, but at that time there was little previous experience to guide the Navy Department, and, lacking this, the details of the installation were left to the Edison Company, who naturally desired to make the installation as successful as possible.

The number and type of lamps actually installed differed slightly from the contract specifications, as there were put on board 173 10-candle-power, 70 16-candle-power, and 4 32-candle-power lamps, a total of 247 lamps and 2,978 candle-power.

The dynamo was of the Edison "L" type, shunt wound with a terminal voltage of 110 and a capacity of 120 ampères, and was rated as a 150 16-candle-power machine. This would make the rated efficiency of the lamps 5.5 watts per candle-power. At that time (1883) lamps were being made with a rated efficiency of 4.6 watts per candle-power, which would give this 13.2 KW. plant a capacity of almost 2,900 candle-power. As a total of 2,978 candle-power was actually installed, the machine would just about care for the total load in the unlikely case of all lamps burning at the same time.

The dynamo was bi-polar, and the four huge field magnets extended to a height of six feet four inches above the deck. These poles were of heavy masses of iron, wound with a great many turns of wire, and the whole covered with cloth for protection. The resistance of the field was in the neighborhood of 40 ohms so that the field current was about 2.75 ampères and the loss in the field .3 KW. The large masses of iron prevented early saturation of the field, and a more powerful machine could be made with a given armature construction and speed. The poles stood on a zinc plate which was bolted to a cast-iron bed plate. On account of the heavy pole pieces used, the weight of the dynamo was 6,000 pounds.

The armature was built up of a large number of very thin disks of soft iron, separated by mica, and strung upon a wooden core. The laminations were firmly bound together by tie bolts and outside of the cylinder so formed were the coils. Of these there was a large number, wound on a modified Hefner-Atteneck system and firmly bound in place by wire bands to prevent the conductors from flying out. The ends were covered with a wrapping of cloth. The commutator was wide and thick and so adapted for long runs. The brushes

were formed of alternating layers of wire and thin strips of copper. They were mounted in a swinging frame to allow of proper adjustment. On an extension of the armature shaft was mounted a fourteen-inch pulley with a nine-inch face. The armature was driven by an eight-and-a-half-inch belt from the fly-wheel of the engine at approximately 900 revolutions per minute.

The bed plate of the dynamo was made with slots which fitted over tracks so that the machine could be moved along the deck to compensate for the stretching of the belt. The dynamo was held down by strong bolts passing through the bed plate and the deck, the bolt holes in the bed plate being elongated so that necessary movement of the machine along the tracks would not be interfered with. In addition to the holding down bolts a heavy brass brace was fitted from the top of the dynamo to the bed plate of the engine. The dynamo occupied a floor space 68 inches by 40 inches.

The engine was a horizontal Armington and Sims. It had a single cylinder 9½ inches in diameter with a 12-inch stroke, and made about 270 revolutions per minute. The engine occupied a floor space 100 inches by 60 inches, and weighed 4,500 pounds. The initial steam pressure seldom exceeded thirty pounds per square inch, and this pressure was not always maintained, especially in port. The engine bed was heavy, solid and stiff, and took the bending stresses of the guides at its upper surface. Two main pillow blocks carried the steel crank shaft and supported the two wheels, one of which was the fly-wheel, and the other the pulley over which the belt to the dynamo ran. The steam cylinder was overhung with the exhaust pipe leading down to the deck. The crank pin was oiled by a "wiper," while sight feed cups supplied the remainder of the lubrication. Oil for the plant was obtained from the Vacuum Oil Company. Mineral oil was used entirely, that for the engine being 600 W Cylinder Oil and that for the bearings Electric Machinery Oil. Both oils cost sixty-

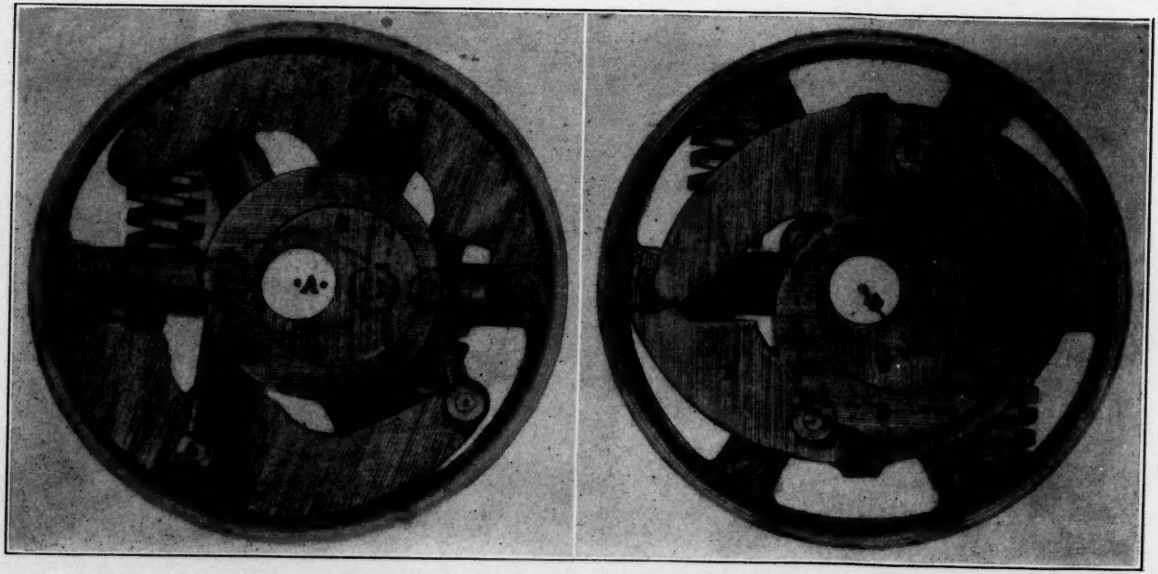
five cents a gallon. It was found unnecessary to use sperm oil on the bearings of the dynamo.

A piston valve with a double port was used with this engine so that approximately boiler pressure was admitted to the cylinder and continued to the point of cut-off with practically no wire-drawing of the steam. This lack of wire-drawing was a very unusual feature in engines at that time. The use of the double ported valve enabled the clearance to be kept down to about 5 per cent. This low clearance was also unusual in an engine where the diameter of the cylinder bore such a large relation to the length of the stroke.

The governor on the Armington and Sims engine was of a novel and ingenious type.

Two weights "1" were carried on and formed a part of the arms pivoted to the governor pulley. The position of these weights as determined by the speed and the action of the springs, determined the position of the eccentrics "C" and "D," and thus the position and motion of the valve and the point of cut-off, flying out and giving a higher rate of expansion as the load on the engine diminished or as the steam pressure rose. The eccentric "C" was set inside the eccentric "D" and connected to the governor arms in such a way that as the weights separated or approached each other, the eccentrics were turned on the shaft so as to cause their throws to coincide or to oppose each other. The action was almost precisely the same as that of a Stephenson link operating between full and mid-gear. The range of action gave very close governing between zero and three-quarters stroke. The action of the governor was tested on board at one time by suddenly throwing off about half load. No perceptible change in speed was noted on the engine.

No figures are available as to the power delivered by this engine on test, but a little speculation is rather interesting. The rated full load on the generator with all lamps burning was about 14 KW. (18.75 H.P.). Assuming 90 per cent efficiency of the generator, which efficiency was obtained on similar machines, the engine had to deliver a little over 20 H.P.



Governor and Eccentrics Minimum Throw.

Governor and Eccentrics Maximum Throw.

If we assume a thirty-pound initial steam pressure, which the ship's officers said they did not always get, an eighteen-inch vacuum, which was about the average obtained, 5 per cent clearance and 20 per cent compression which this type of engine usually used, and cut off at 50 per cent of the stroke, the theoretical mean effective pressure will be a trifle over thirty-one pounds. If we assume a diagram factor of .7, which is attained on small fast-running engines of this type, the mean effective pressure was a trifle less than 22 pounds.

At 270 r.p.m. the horsepower developed equaled

$$\frac{2 \times 270 \times 22 \times 70.88}{33{,}000} = 25 +, \text{H.P.}$$

Thus there was ample power to carry the full load of the generator even with the engine cutting off at half stroke.

The installation was made under the supervision of Mr. Miller F. Moore, formerly an engineer in the Navy, and at that time General Manager of the Edison Company. Carpenters were used to run the wire in as they were needed to make a good job on the moulding.

The good design and workmanship of the engine and the care it received on board the *Trenton* are indicated by the fact that during fifteen months' use the only renewal necessary was a set of crank-pin brasses.

The engine and dynamo belted together occupied a floor space of 101 square feet, and weighed ten thousand, five hundred pounds. It is interesting to note that the 500 KW. turbo-generators to be installed on the battle cruisers will take up a floor space of but 106.5 square feet, although they will weigh forty-six thousand pounds. The advance of electrical engineering is well shown in this development from 15 KW. to 500 KW. in practically the same floor space.

The switchboard for the *Trenton's* lighting plant was located near the bulkhead in the after end of the dynamo room. It was massively built of slate with the bus bars on the front,

and on it were mounted the circuit switches for the various lighting circuits of the vessel. No voltmeter or ammeter was fitted to the board, as the development of these instruments had not progressed to the point where they would operate satisfactorily aboard ship. The voltage at the lamps was regulated by the eye, being adjusted to give less candle-power than their rating in order to prolong their life. The 16-candle-power lamps were adjusted to give 12-candle-power as closely as could be determined. The efficiency of the lamps was not considered, as the predominant desire with this installation was to show reliability as compared with oil lamps, and long life of the lamps was secured by burning them at a low candle-power. Even 12-candle-power lamps as compared with oil lamps were such an improvement that apparently no objection was raised to the lower rating.

An automatic voltage regulator was furnished with the dynamo, but its use was discontinued after six months. This regulator consisted of a rheostat in the field circuit which was automatically shifted by a solenoid when the current in the external circuit rose or fell due to an increase in the number of lamps in use. As the generator was a shunt machine, additional current in the external circuit would cause the voltage to fall off. With the automatic regulator, additional current would move the solenoid, cutting out field resistance, and by allowing additional current to flow in the field, bring the voltage up again.

Reports from the ship indicated a belief that failure of lamps was due to the use of this regulator. No explanation was offered for this belief, but it may be supposed that the regulator failed to reduce the voltage when the load was decreased. This excess voltage would tend to decrease the life of the lamps considerably.

The switchboard was fitted with switches for the following circuits: spar deck, gun deck, berth deck, orlop deck, wardroom and admiral's quarters, fire and engine rooms and holds.

These main circuits were of number 12 or 14 single conductor cable (6,530 or 4,107 circular mils). They were led about the ship in pine mouldings. These mouldings consisted of battens with two grooves for the mains over which a cap was fastened with screws after the wires were run. When well painted, this moulding was fairly watertight.

For the branch wires, number 18 (1,624 circular mil) wire was used. This was run in a smaller moulding with grooves, which was screwed to beams and bulkheads. Much trouble was experienced with salt water getting behind this moulding and rotting the insulation of the branch leads.

The wire for the mains was a single strand, tinned and covered with rubber insulation. That for the branch leads was for the most part untinned and covered with cotton insulation and painted. A small number of the branch leads were tinned and insulated with rubber, which later proved to be of very inferior quality. The wire for the portable deck lanterns and engine-room portables was twin conductor, two fine flexible wires being surrounded by gutta percha covered with jute braiding. This wire was very unsatisfactory as the gutta percha hardened and became brittle in cold weather so that it cracked off, and in warm weather softened sufficiently to allow the wires to come into contact with each other and cause a short circuit.

The mains were fitted with numerous cut-out boxes from which three or four lamps were controlled. The boxes were fitted with fuses to protect the mains.

More trouble was experienced from defective wiring on the *Trenton* than from any other part of the electric plant. Salt water was as great an enemy to her simple plant as it is to our complicated systems of to-day. Wires were frequently found entirely corroded through from the action of the salt water and the copper reduced to a greenish powder. The resistance of this powder was high enough so that with the ordinary currents flowing which would not blow a fuse, enough heat would be generated to set the moulding on fire.

The Edison lamps were fitted about the ship in ceiling fixtures, bulkhead standing lights, electroliers, and bracket fixtures, very much as they are to-day. Reading lamps and a cigar lighter were furnished for the wardroom. Portables were furnished for use on deck, in staterooms, and in the engineer department. Running lights, anchor lights and gangway lights were fitted. The running lights had two lamps in each box, fitted with a relay which would cut in the second lamp if the first one failed. This automatic feature was not a success, however.

A night signalling set was devised, using deck lanterns. Two lamps were hung a certain distance apart, depending on the distance of the ship to which the signal was made, and connected to a keyboard with two keys. By pressing one key, one lamp was lighted, and by pressing the second key, both lamps were shown. By means of the Myer code, signals could be sent accurately and rapidly.

An illuminating set for the quarter deck was constructed by stringing one hundred sockets on a cable. This set was tried out on July 4, 1884, and contributed materially to the brilliancy of the occasion.

A deep-sea lamp was fitted up with eighty feet of cable for use over the side.

The lamp sockets were secured to the fixtures by set screws, as were the shade holders. They frequently worked loose, due to vibration. One wire inside the lamp socket was soldered into the ring into which the lamp base screwed. This soldered joint frequently carried away.

On the trip across the Atlantic soon after the lighting set was installed very heavy weather was encountered and the ship worked a great deal. This caused considerable trouble with the wiring, but the fuses prevented serious damage.

Many of the lamp sockets were attached to the fixtures with spiral springs, and the mortality of lamps in these sockets was much less than in those rigidly screwed to the fixtures.

Many lamps gave a life of over five thousand hours. Much of this longevity was due to operating the lamps at a lower candle-power than their rating; life being obtained at the expense of efficiency. At first the expenditures of lamps was very high, but it was gradually reduced as witness the expenditure during the year 1884:

Lamps expended first quarter 1884.....	334
Lamps expended second quarter 1884.....	78
Lamps expended third quarter 1884.....	65
Lamps expended fourth quarter 1884.....	50

Lamps were lost from various causes. Of the last fifty lamps expended, five were of defective construction such as bad connection of the carbon filament with the copper wire inside the lamp, and broken wires inside the plaster of Paris; fifteen were accidentally broken; and remainder broke down from short circuits and inherent defects which shortened their life.

The objection to electric lighting on account of its not standing the shock of gun fire proved ill-founded. During the first quarter when target practices were held not a lamp or shade of the electric plant was removed, and but one lamp was broken. This lamp was eighteen inches from the breech of a gun when run in and was not fitted with a spring fixture.

The successful operation of the electric plant on the *Trenton* was due to the constant supervision of the officers in charge of the equipment. Lieut.-Commander Bradford, who supervised the installation of the plant and who was in charge at first, and Ensign E. H. Tillman (now Comdr., U. S. N., Retired), who succeeded him, were indefatigable in their efforts to keep the plant in efficient condition. They made many valuable reports which formed the basis of specifications for the generators, wiring and lamps for subsequent electric plants.

The lighting of men-of-war by electricity was proven practicable, and the experiment was such a success that the question was settled in our service.

The *Trenton* was re-wired in 1887 at the end of her cruise, great attention being paid to the insulation of the circuits. Thirty-nine new lamps were added, and the fixtures repaired and re-polished. The dynamo and engine were found in excellent condition.

The plant continued in successful operation until the *Trenton* was lost in Samoa in March, 1889, the dynamo continuing to run until the fires under the boilers were extinguished by the incoming water.

THE RADOJET AIR PUMP.

By M. C. STUART, ASSOCIATE, AND ARTHUR H. SENNER.

The use of the steam ejector type of air pump in the United States Navy and the Merchant Marine Service of this country is becoming increasingly general.

Before the war the use of the ejector air pump in connection with a condenser was regarded with some doubt; in fact, so great was the feeling of uncertainty regarding the dependability of this extremely simple innovation, that very often when the ejector air pump was accepted for use it was installed alongside of a reciprocating air pump which was considered more dependable. In most installations of this kind, however, the steam ejector pump proved to be entirely satisfactory.

During the war when manufacturers could not meet the increasing demands for marine auxiliaries, the steam ejector air pump was substituted quite extensively for the reciprocating air pump which required more time and material for its construction. The service which the ejector air pump gave, due to its reliability, simplicity and compactness removed almost entirely any prejudiced ideas which had previously existed concerning its practicability. This type of air pump is now accepted as standard equipment in the United States Navy, and, in fact, is the only air pump considered in the latest capital ships.

Since 1915, a number of steam air ejectors of various types have been tested exhaustively at the U. S. Naval Engineering Experiment Station, Annapolis, Md., under authorization of the Bureau of Engineering of the Navy Department. These tests show that the various improvements which have been made from time to time by several manufacturers have re-

sulted in steadily increasing efficiencies. Herein are given descriptions and results of tests of two Radojet Air Pumps manufactured by the C. H. Wheeler Manufacturing Company, which show high efficiencies.

DESCRIPTION OF PUMPS.

The steam ejector type air pump is a compressor in which the fluid to be compressed is entrained by one or more jets of steam, which move through the entrainment space at a high velocity. The entrainment of the fluid to be removed is accomplished by the surface friction of the steam jets and surrounding fluid. The kinetic energy supplied by the steam jets, and contained in the mixture after entrainment, is converted into pressure in a channel called the diffuser. The fluid to be removed from a condenser is air saturated with water vapor.

Both of the pumps tested were of the two stage type, and similar in general design, differing in only one principal feature; one had an intercondenser between the first and second stages, while the other was of the plain two stage type.

The external appearances of the pumps are shown in the photographs, Figures 1 and 3. Cross-sectional views of the pumps are shown in Figures 2 and 4. Referring to Figure 2, the view of the inter-condenser Radojet, the following description explains the various parts and their particular functions. This ejector air pump consists essentially of two steam jets arranged in series, and with a condenser interposed. The work of compression of the fluid from the condenser is divided in unequal parts between the two jets or stages. The ejector is intended to remove only the air and non-condensible vapor; the condensed steam being removed by a separate condensate pump.

Referring to Figure 2, the operation is as follows: The jet in the first stage results from the expansion of the steam through 19 convergent-divergent nozzles N_1. The air to be compressed enters at S into the suction chamber of the first stage, is then entrained by the steam jet, emerging from the

FIG. 1.—PHOTOGRAPH OF INTERCONDENSER RADOJET.

FIG. 3.—PHOTOGRAPH OF RADOJET WITHOUT INTERCONDENSER.

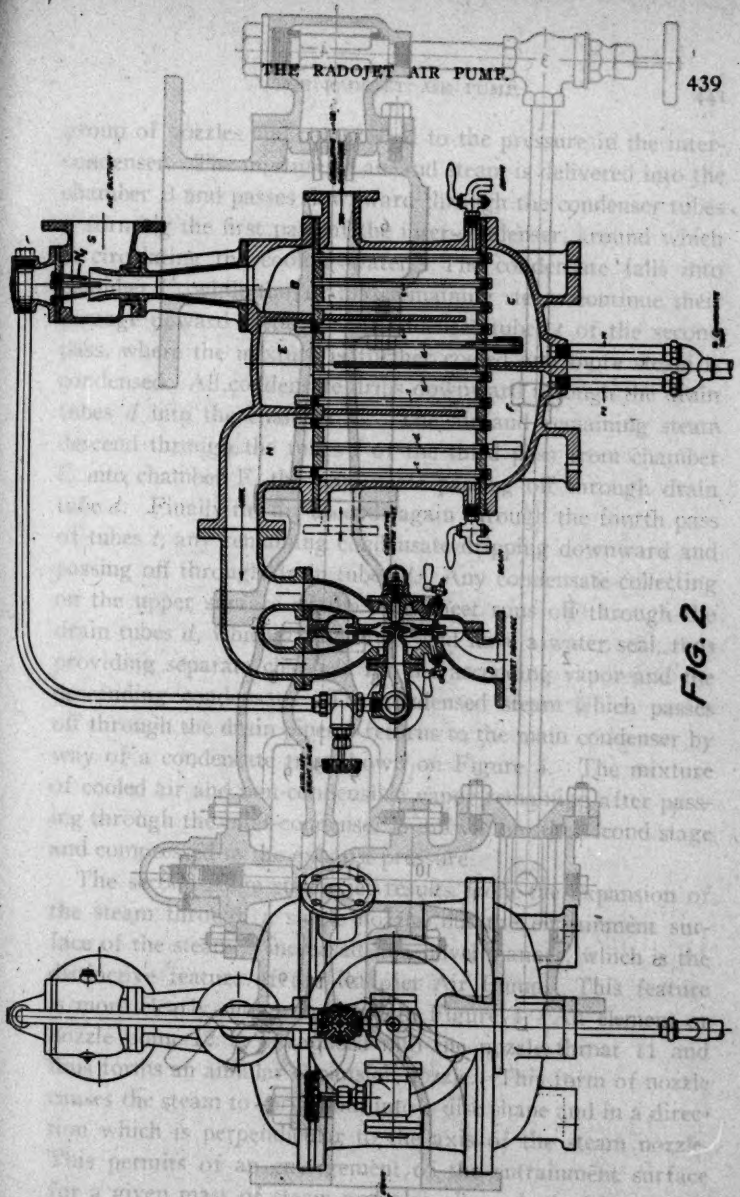


FIG. 2.—SECTIONAL VIEW OF INTERCONDENSER RADOJET.

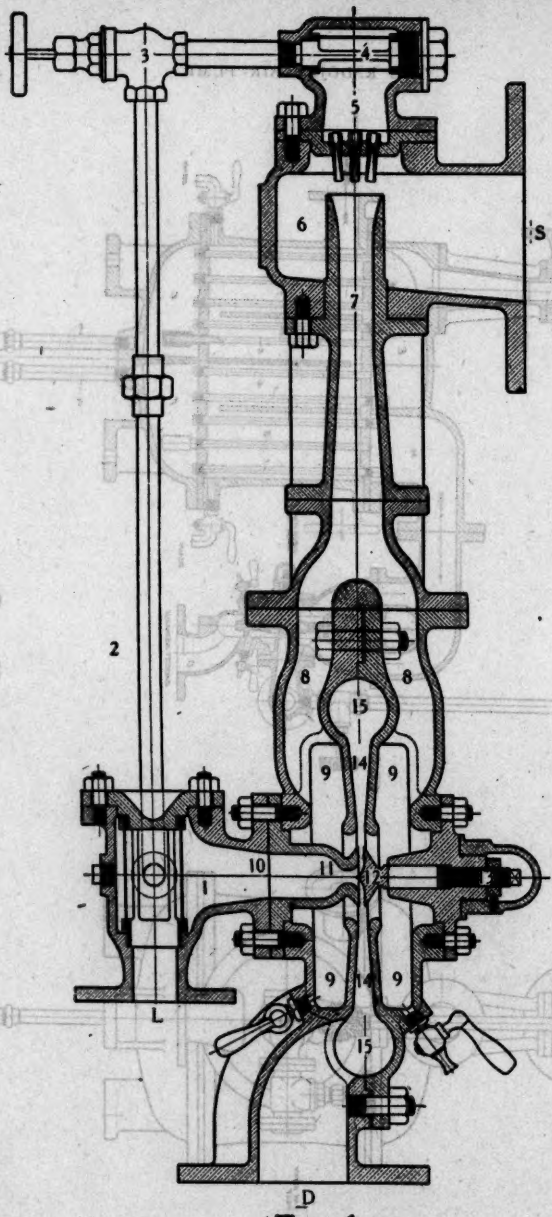


FIG. 4

Cross-Sectional View of the Radojet Vacuum Pump.

FIG. 4.—SECTIONAL VIEW OF RADOJET WITHOUT INTERCONDENSER.

group of nozzles and compressed to the pressure in the inter-condenser. The mixture of air and steam is delivered into the chamber B and passes downward through the condenser tubes *t*, forming the first pass of the inter-condenser, around which is circulating the cooling water. The condensate falls into chamber C, while the air and remaining steam continue their passage upward through the condenser tubes *t* of the second pass, where the mixture is further cooled and more steam is condensed. All condensate drips downward through the drain tubes *d* into the chamber C. The air and remaining steam descend through the tubes *t* of the third pass from chamber E into chamber F, the condensate passing off through drain tube *d*. Finally the air ascends again through the fourth pass of tubes *t*, any remaining condensate dripping downward and passing off through drain tubes *d*. Any condensate collecting on the upper surface of the tube sheet runs off through the drain tubes *d*, which at the lower end have a water seal, thus providing separate channels for the ascending vapor and the descending condensate. The condensed steam which passes off through the drain pipes *p* returns to the main condenser by way of a condensate trap shown on Figure 5. The mixture of cooled air and non-condensible vapor remaining after passing through the inter-condenser is drawn into the second stage and compressed to the exhaust pressure.

The second stage steam jet results from the expansion of the steam through a single nozzle, but the entrainment surface of the steam is increased in a novel manner, which is the distinctive feature of the Radojet Air Pump. This feature is more clearly shown in detail in Figure 4. An element or nozzle point 12 is introduced into the nozzle throat 11 and thus forms an annular expansion nozzle. This form of nozzle causes the steam to spread out into a disc shape and in a direction which is perpendicular to the axis of the steam nozzle. This permits of an enlargement of the entrainment surface for a given mass of steam and also allows both sides of this

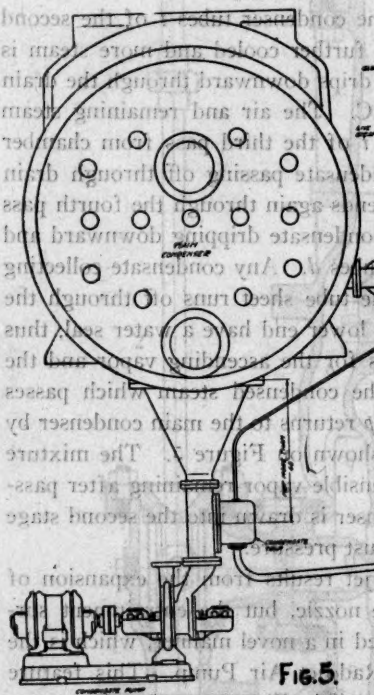


FIG. 5.

FIG. 5.—INSTALLATION OF INTERCONDENSER RADOJET.

disc-like jet of steam to entrain the air in passing across the second stage suction chamber 9. The steam and entrained air then pass into the annular diffuser 14, thereby compressing the mixture to the exhaust pressure. The mixture of air and steam discharges into the casing 15, which has the discharge opening D. The mixture discharged at D may be delivered into a small tank supplied with fresh water for boiler feed, as diagrammatically illustrated in Figure 6.

The method of installing this ejector on board ship is shown in Figure 5. The ejector is connected to the condenser so that it will withdraw the air and the vapor which is noncondensible at the temperature and pressure existing at the pump suction of the condenser. A separate condensate pump is provided to remove the condensed steam from the condenser. The ejector discharges into a "Feed and Filter Tank" shown on Figure 6; thus the heat value of the steam from the second stage nozzle is utilized by heating the feed-water. It is also important to note that by circulating the condensate from the main condenser around the inter-condenser tubes the heat liberated by the first stage steam in being condensed is taken up by the condensate and thus saved. Thus it is seen that with suitable installation all the heat in the steam supplied to the ejector which is not utilized in performing the useful work of compression of the fluid may be saved by the economizing method.

The Radojet Air Pump without inter-condenser as shown in Figure 4 is very similar in design to the inter-condenser Radojet Air Pump described above and the one description will serve for both pumps with the omission of the inter-condenser description.

METHOD OF INSTALLATION ON BOARD SHIP.

The arrangement of either Radojet Air Pump for marine surface condensers is diagrammatically illustrated in Figure 6. This figure shows the feed and filter tank, previously referred to, which receives both the condensate from the main con-

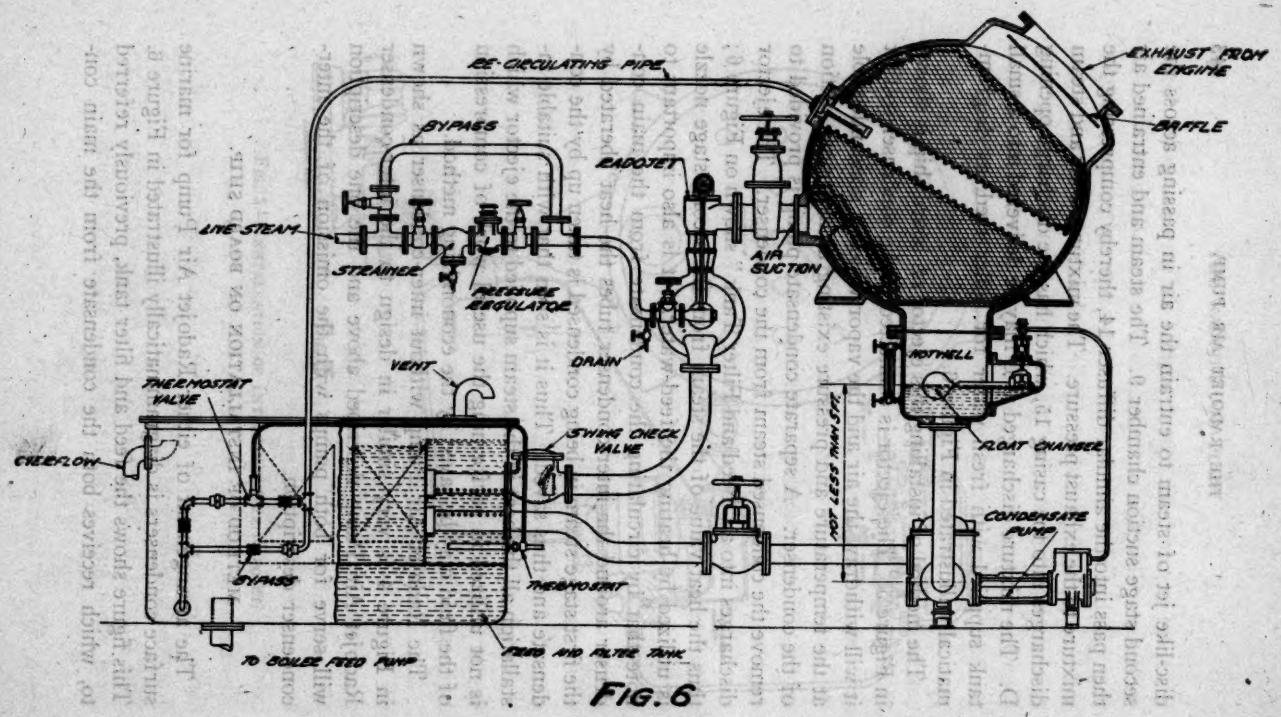


FIG. 6

FIG. 6.—INSTALLATION OF RADOJET WITHOUT INTERCONDENSER.

denser, and the discharge from the air ejector. Under ordinary operating conditions, the mixing of the condensate and air pump discharge produces a certain water temperature, depending upon the amount of condensate available. However, the continued operation of the ejector when the main engines or turbines are stopped would result in excessive heating and boiling of the water in the mixing tank. To prevent this, an automatic thermal-control valve is used which, when the temperature in the mixing tank exceeds a certain value, causes the hot water to be drawn from the mixing tank into the condenser, where it is cooled and returned by the condensate pump to the feed and filter tank, thus preventing any excessive rise in temperature when the exhaust from the main engine supplies insufficient condensate.

GENERAL ARRANGEMENT FOR TEST.

In Figure 7 is shown the general arrangement for test. The method used for testing the ejector was developed to embrace the following features:

(*a*) Exact duplication of condenser conditions at air ejector entrance.

(*b*) Positive control of air quantity.

(*c*) Exact measurement of air quantity.

(*d*) Positive and convenient control of temperature and air-moisture ratio of mixture entering ejector.

With the arrangement used, no air could possibly be handled by the ejector, without its having entered by way of the standard air orifice shown on drawing; also it is possible either to supply dry expanded atmospheric air to the ejector, or the expanded atmospheric air may be saturated with water vapor at any desired temperature before entering the ejector.

The steam line to the ejector is shown on this figure. The pressure and temperature of steam are measured at each stage. Calorimeter samples are taken from the steam line just ahead of the valves controlling the pressures at the two stages. In

FIG. 6.—INSTALLATION OF RADOJET WITHOUT INTERCONDENSER.

order to prevent the accumulation of foreign matter in the nozzle area, strainers were placed in the line as shown in sketch.

The air is controlled and saturated with moisture as follows: The atmospheric air enters the humidifier through a standard calibrated air orifice of a given capacity. The air is then divided into fine streams by passing through a perforated disc placed in the line and held between two flanges as shown on the sketch. This dividing of the air into streams produces closer mingling of the air with the water in the humidifier, thus insuring its becoming saturated with vapor. The humidifier which contained water from one-fourth to one-third of its height was replenished from the "water make-up tank for humidifier" shown on sketch.

An essential feature of the method is the ability to maintain the mixture of air and water at a constant temperature or to vary it as may be desired. This is accomplished by admitting steam into the humidifier below the water level as shown on sketch.

Upon leaving the humidifier the air is moisture saturated and possibly contains in suspension moisture droplets which should leave the mixture upon entering the separator. In order to control and determine the properties of the saturated mixture which was drawn into the ejector, measurements were made of the pressure and temperature in the separator, which, as is shown clearly on the sketch, is the first stage suction space. A floating scale, single tube mercury column and two thermometers, the exposed bulbs of which were inserted directly into the separator space, were employed for measuring the pressure and temperature, respectively, of the mixture.

When no air was admitted to the ejector, that is, when a blank was put in place of the calibrated air orifice, but the water in humidifier was being heated by the admission of steam into the water, the pressure at the pump corresponded to the temperature of saturated vapor. Thus it was learned

that no superheat existed in the system when no air was admitted. Now, when air was admitted and the water in humidifier heated as before, the total pressure was higher than that corresponding to the temperature at the pump, for saturated steam; the additional pressure being due to the partial air pressure. The condition of the mixture of air and vapor required to produce a saturated mixture, was verified by another criterion, namely, the agreement between the amounts of water admitted to the humidifier and that required to saturate the dry air passing through the system at the existing temperature and pressure. Now, from the steam nozzle calibration the rate of steam flow was known, and any additional amount of condensate from the ejector must necessarily result from moisture which was introduced at the humidifier to saturate the air at the existing pressure and temperature. With the aid of Figure 30, comparison was made between the excess weights gotten and the amount of water required to saturate the air, and the comparison in all cases indicated that a condition of saturation existed at the pump suction. In fact, at times excess moisture in the form of droplets suspended in the mixture was carried into the pump without, however, any effect upon the performance at the pump, as will be shown later.

As is shown on Figure 7, one exposed bulb thermometer was placed in the line entering the separator, and the other was placed directly at the pump suction pipe; both thermometers registered practically the same temperature and thus indicated a state of uniform temperature in the system.

The saturated air is compressed in the first stage and discharged at a certain pressure to the suction space of the second stage. As is shown on sketch, the pressure at the second stage suction was measured by means of a mercury column. In the second stage, the mixture is compressed to the exhaust pressure; the back pressure being regulated by a valve placed in the exhaust line. The exhaust was conducted to a small con-

denser, where the steam discharged from the second stage was condensed, after which it was delivered to weighing tanks. The temperature of the exhaust is measured as shown on sketch.

Dry or expanded atmospheric air may be supplied to the ejector by removing all water from the system, that is, the humidifier and separator, and allowing air to enter through the calibrated orifice.

Hygrometric data were obtained from wet and dry bulb thermometers, placed close enough to the air nozzle entrance to obtain fair samples of the air entering the pump.

DISCUSSION OF RESULTS.

Figures 8 and 9 are tables showing the data and results from the tests on the inter-condenser Radojet. On Table 1, Figure 8, are tabulated, the runs made with the ejector handling dry expanded atmospheric air. On Table II, Figure 9, are tabulated the runs made with the ejector handling moisture saturated air. Information of importance and interest is given by runs 53 and 54, Figure 9. These runs were made to determine if a moderate quantity of excess moisture in the form of droplets in suspension would affect the vacuum maintained by the pump, other conditions remaining constant. It is highly probable that in the case of a dry-air pump operating in conjunction with a condenser, the mixture drawn into the pump contains moisture droplets in suspension, because of the improbability of perfect separator action within the condenser. By comparison with the performance of the pump on run No. 52, in which the pump handled the same amount of fluid but without any excess moisture droplets, it is learned that the performance is not impaired by moderate amounts of moisture in the form of droplets.

Figure 10 contains the steam flow curves of the inter-condenser Radojet, as would be expected from the law of flow through nozzles, the flow is proportional to the pressure. At the operating pressure of 100 pounds gage, the steam flow is

204 pounds per hour in the first stage and 603 pounds per hour in the second stage, a total of 857 pounds per hour.

Curves showing capacity and steam economy when handling dry expanded atmospheric air are given on Figure 11. The small numbers at the test points correspond to the reference numbers on Table I, Figure 8. The operation of the ejector with dry expanded atmospheric air has no direct bearing on the operation of the ejector as used with a condenser, where it handles moisture saturated air. The results are given because frequently tests are reported upon this basis, although, as will be shown later, the actual performance of the ejector on a condenser may be obtained only by means of a test with moisture saturated air.

On Figure 12 are to be found curves of capacity and efficiency when handling dry expanded atmospheric air based on the heat supplied in the steam above the pressure of the exhaust, and efficiency based on the energy supplied through adiabatic (isentropic) expansion from nozzle pressure to exhaust pressure. The work theoretically required is based upon isothermal compression of the air; this kind of compression having been adopted as a standard in order to permit of a comparison with the work in a general type of air compressor in which the minimum possible amount of theoretical work required is that resulting from isothermal compression.

Figure 13 gives the intercondenser Radojet characteristics when handling moisture saturated air. Each curve represents a constant quantity of dry expanded atmospheric air, which, subsequent to its expansion through the air-measuring orifice had been saturated with water vapor before reaching the pump suction. At each constant air quantity the temperature of the saturated mixture was varied from 60 to 120 degrees Fahrenheit. The first curve at the left gives the absolute pressure at the pump when no air was admitted into the system, and the ejector was allowed to handle only the dry saturated steam arising from the water contained in the humidifier. As is the

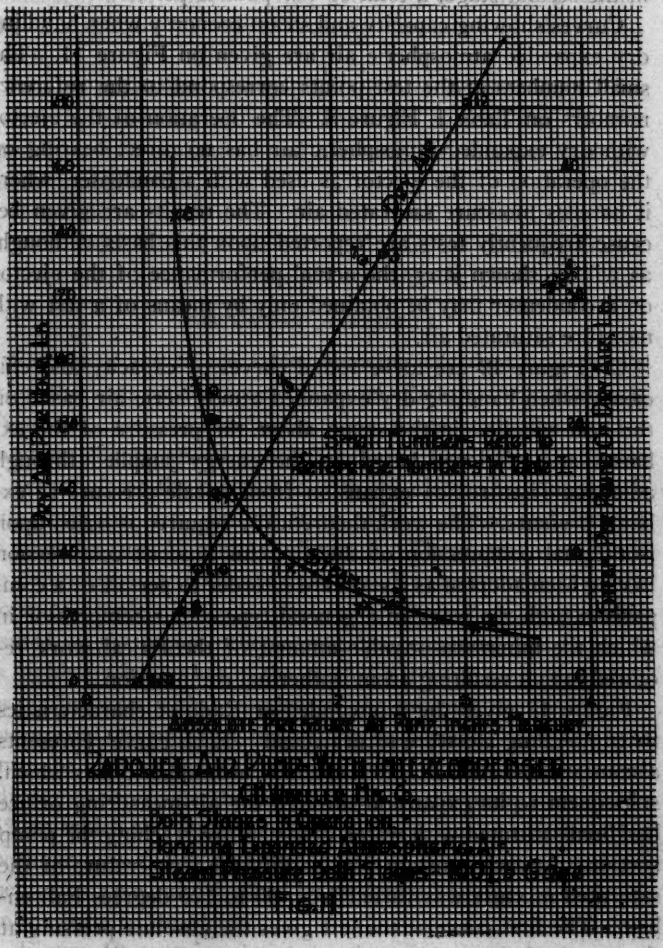


FIG. 11.—CHARACTERISTICS OF INTERCONDENSER RADOJET HANDLING DRY AIR.

case in all pumps of the steam ejector type, the pressure along this line corresponds to the temperature of saturation for the vapor. The shape of the curves of Figure 13 being concave toward the horizontal axis and practically parallel to or at least having the general shape and direction of the first curve at the

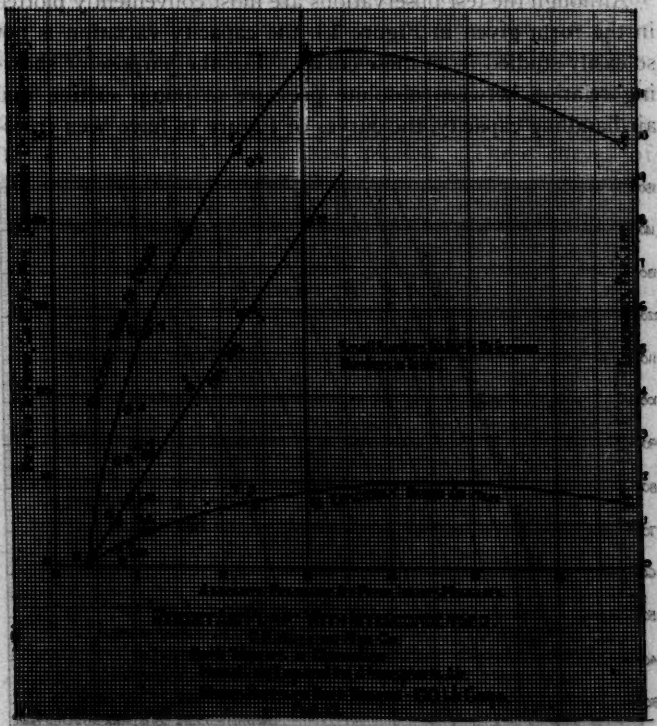


FIG. 12.—CHARACTERISTICS OF INTERCONDENSER RADOJET HANDLING DRY AIR.

left, is obvious upon the following consideration. The curve at the left represents the pump characteristic when handling no air, and the temperature corresponds to the pressure of saturated steam. Now, as the temperature increases the corresponding pressure of saturated steam increases and when

plotted gives the curve as shown. At a given temperature of mixture at pump, the horizontal distance between the curve of no air in mixture, or the first curve at the left, and any other curve representing performance with air saturated with vapor, is the partial air pressure due to the air in the mixture.

Although the test observations are most conveniently plotted in the form given in Figure 13, the capacity variation is not so easily studied from these curves. For the purpose of showing relation between absolute pressure at pump suction and air handling capacity, the curves of Figure 14 have been cross-

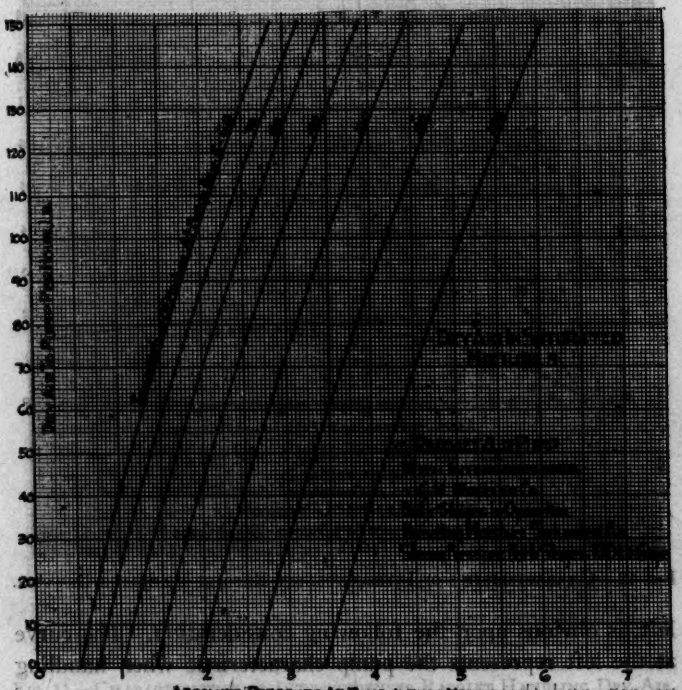


FIG. 14.—CHARACTERISTICS OF INTERCONDENSER RADOJET—HANDLING MOISTURE SATURATED AIR.

plotted from Figure 13. On this figure each curve gives the pressure at pump *vs.* weight of dry air in the mixture of moisture saturated air, with a constant temperature of mixture at pump suction. Each of these curves intersects the axis of absolute pressure at pump (the ordinate of zero air quantity) at a pressure corresponding to the constant temperature for each curve, which is the partial vapor pressure, the partial air pressure being zero, since there is no air present at these points.

In order to show the variation in partial air pressure at various quantities of dry air in the saturated mixture, the curves of Figure 15 have been plotted. These curves of

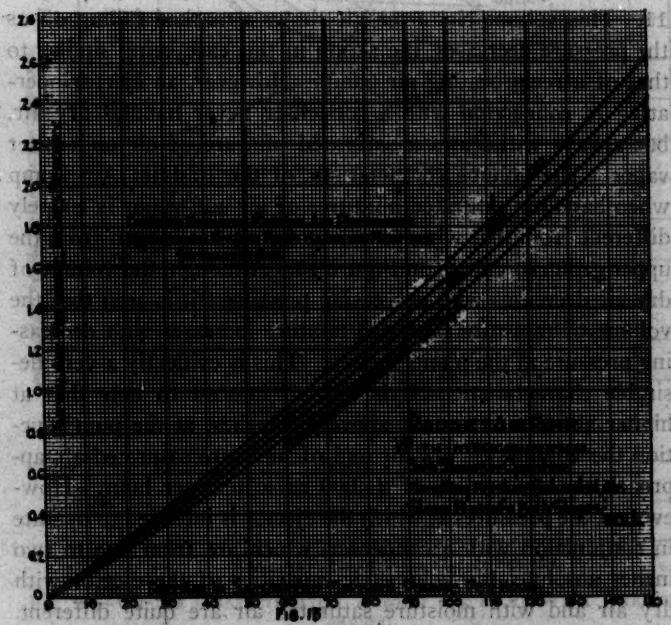


FIG. 15.—PARTIAL AIR PRESSURES MAINTAINED WHEN HANDLING A MIXTURE OF MOISTURE SATURATED AIR.

weight of dry air in saturated mixture *vs.* partial air pressure, were obtained directly by plotting the partial air pressures along the constant temperature lines of Figure 14 against the dry air to pump in pounds per hour.

VOLUME CHARACTERISTICS.

Although the function of an ejector is to remove a certain weight of air, a knowledge of its volume characteristic is very valuable for certain purposes. To this end Figure 16 has been prepared. The volume of dry air handled has been computed from the dry air characteristic of the ejector, plotted on Figure 11. This volume characteristic shows a marked falling off as the pressure decreases, the action in this case being similar to that of a reciprocating air pump. However, an ejector operating in connection with a condenser never handles dry air, but always withdraws a saturated mixture of air and water vapor. The volume characteristic of the Radojet Air Pump when handling saturated mixtures of air and vapor is entirely different from that when handling dry air, as is seen from the upper group of curves on Figure 16. Although the volume of dry air decreases with decreasing pressures, it is seen that the volume of moisture saturated mixture increases with decreasing pressure at the pump suction. This is evidently a very desirable characteristic. It is also of interest to note that at higher absolute pressures, above 4.5 inches, at the pump suction the volumes of dry air and of moisture saturated air approach the same value of 24,000 cubic feet per hour. However, at the pressures at which the ejector is intended to operate in conjunction with a condenser, which are from one to two inches absolute, the volumetric capacities of the ejector with dry air and with moisture saturated air are quite different. This shows the error of the common assumption that a steam ejector air pump handles equal volumes of moisture saturated air and dry air at the same pressure at pump suction.

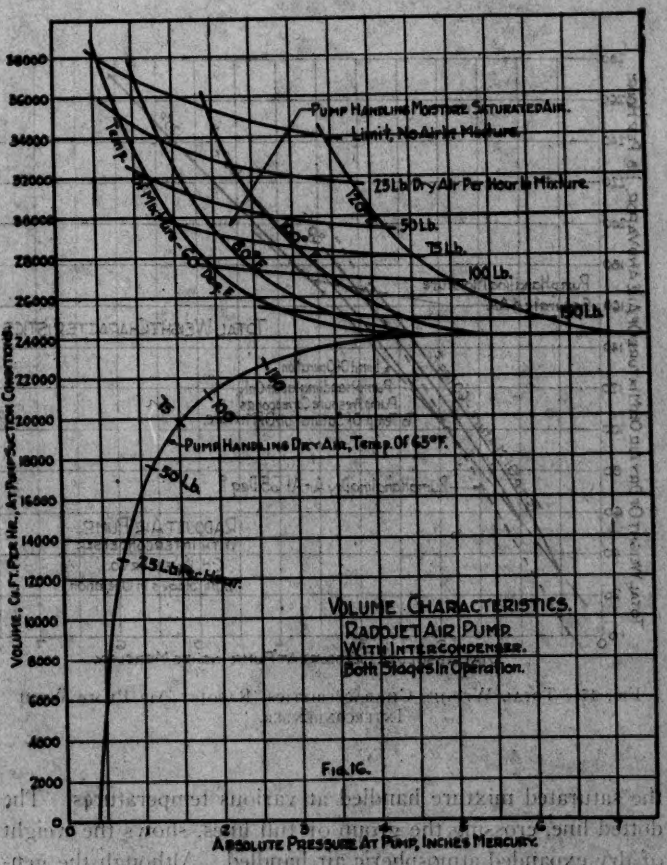


FIG. 16.—VOLUME CHARACTERISTICS, RADOJET AIR PUMP WITH INTERCONDENSER.

A comparison of the weight of expanded atmospheric air and vapor saturated air handled is also of interest. On Figure 17, the full lines show total absolute pressure *vs.* weight of

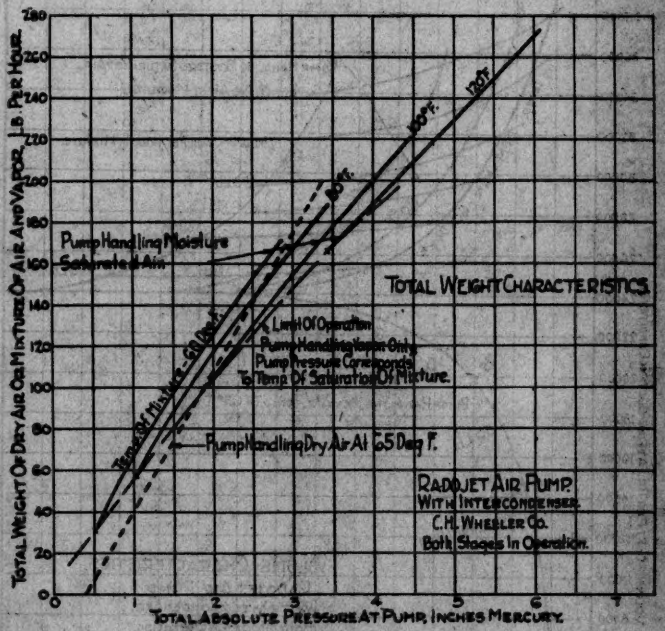

Total Weight Of Dry Air Or Mixture Of Air And Vapor, Lb. Per Hour.
Total Absolute Pressure At Pump, Inches Mercury.
Pump Handling Moisture Saturated Air
Total Weight Characteristics
Limit Of Operation Pump Handling Vapor Only Pump Pressure Corresponds To Temp. Of Saturation Of Mixture.
Temp. Of This Mix. - 60 Deg. F.
Pump Handling Dry Air At 65 Deg. F.
Radojet Air Pump With Intercondenser. C. H. Wheeler Co. Both Stages In Operation.
100°
120°
140°

FIG. 17.—TOTAL WEIGHT CHARACTERISTICS, RADOJET AIR PUMP WITH INTERCONDENSER.

the saturated mixture handled at various temperatures. The dotted line, crossing the group of full lines, shows the weight of dry expanded atmospheric air handled. Although the general shape of the characteristics are the same, under certain conditions the weight of the saturated mixture is greater and under other conditions it is less than the weight of expanded atmospheric air.

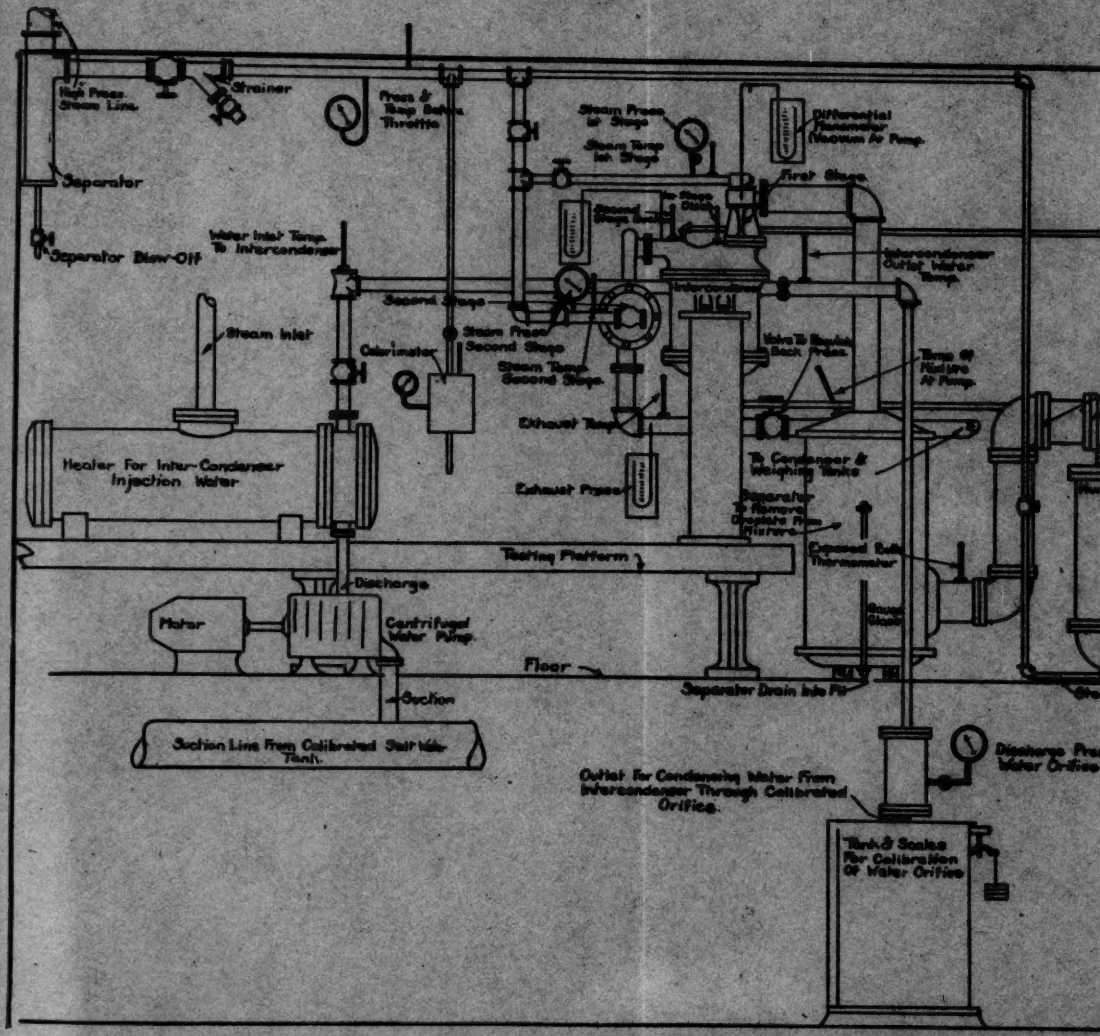
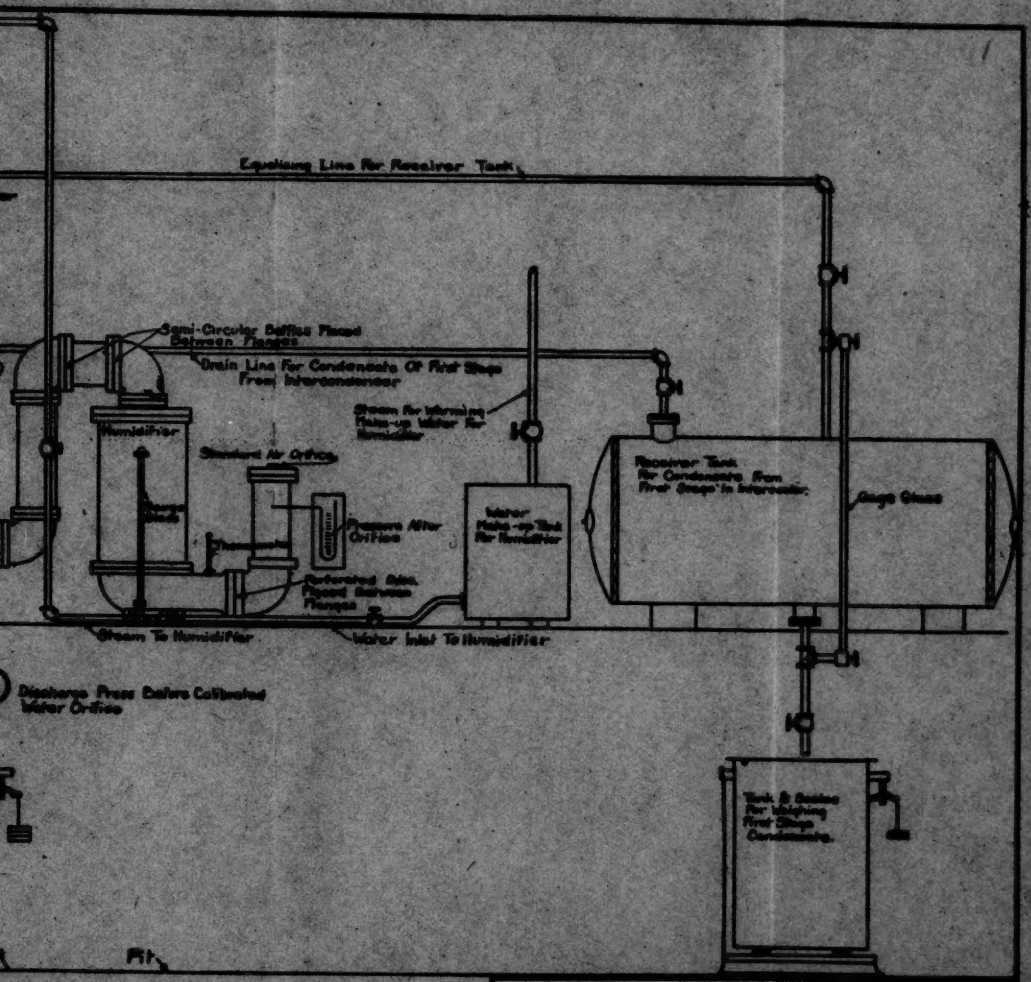


Fig. 7

Fig. 7.—Sketch of Test Arrange


Equalising Line For Receiver Tank
Semi-Circular Baffles Placed Between Flanges
Drain Line For Condensate Of First Stage From Intercondenser
Steam for Warming Make-up Water for Humidifier
Humidifier
Standard Air Orifice
Pressure After Orifice
Water Make-up Tank for Humidifier
Receiver Tank for Condensate from First Stage in Intercooler
Gauge Glass
Reinforced Rubber Pipe Placed Between Flanges
Steam To Humidifier
Water Inlet To Humidifier
Discharge From Entire Calibrated Water Orifice
Tank & Scales for Weighing First Stage Condensate
Fit

ARRANGEMENT FOR TEST OF RADOJET AIR PUMP WITH SURFACE INTERCONDENSER

TEST ARRANGEMENT.

Reference No.	Date, 1920	Duration of Run, Minutes	Room Temperature, Degrees Fahr.	Barometric Pressure, Inches, Mercury	Wet Bulb Temperature, Degrees Fahr.	Water Vapor Per Pound of Dry Air, Lb.	Steam Pressure Before Throttle, Lb. per Sq.In.-Gage	Steam Temperature Before Throttle, Deg. Fahr.	Superheat Before Throttle, Degrees Fahr.	Steam Pressure At Second Stage, Lb. Per Sq. In.-Gage	Steam Pressure At First Stage, Lb. Per Sq. In.-Gage	Steam Temperature At Second Stage, Deg. Fahr.	Steam Temperature At First Stage, Deg. Fahr.	Throttling Calorimeter Pressure, Lb. Per Sq. In.-Gage	Throttling Calorimeter Temperature, Deg. Fahr.	Total Heat In Steam, B.t.u. Per Pound	Steam Quality At Nozzle Conditions, Per Cent	Steam Quality Referred To 264.7 Lb. Per Sq. In., Per Cent	Exhaust Pressure, Lb. Per Sq. In.-Gage	Corresponding Saturated Steam Temp., Deg. Fahr.	Exhaust Temperature By Thermometer, Degrees Fahr.	Superheat At Exhaust, Degrees Fahr.	Heat In Steam Above Temp. Corresponding To Exhaust Pressure, B.t.u. Per Lb.	Energy Available By Adiabatic Expansion, B.t.u. Per Lb.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25
1	Jan. 2	15	58.1	29.92	45.7	0.003983	213.0	393.6	0.0	100.0	99.4	351.9	343.3	5.6	288.0	1184.0	99.5	97.8	0.15	212.5	277.1	64.6	1003.5	149.6
2	Jan. 2	15	58.5	29.92	45.8	0.003650	218.1	395.4	0.0	100.0	100.0	351.9	344.2	6.0	286.8	1184.0	99.6	97.9	0.15	212.5	273.6	61.1	1003.5	149.0
3	Jan. 3	15	50.3	30.33	43.1	0.004108	243.2	404.3	0.0	100.0	101.3	354.9	346.4	4.6	286.6	1184.0	99.4	97.8	0.18	214.3	269.7	55.4	1001.7	147.4
4	Jan. 3	15	49.5	30.32	46.6	0.005915	243.2	404.1	0.0	114.7	114.5	358.9	350.8	4.6	286.6	1184.0	99.2	97.6	1.44	214.1	259.7	45.6	1001.9	155.8
5	Jan. 3	15	58.1	30.22	46.9	0.004150	253.7	408.6	0.0	99.8	100.0	355.0	346.0	4.3	282.5	1187.0	99.8	98.1	0.10	212.7	256.2	43.5	1000.3	149.8
6	Jan. 3	15	54.7	30.22	53.6	0.006400	254.2	408.6	0.0	100.0	101.0	355.2	346.2	4.4	282.8	1187.0	99.8	98.1	0.64	214.3	256.8	42.5	1001.7	148.4
7	Jan. 3	15	57.5	30.21	48.4	0.005055	254.5	407.8	0.0	100.0	101.0	355.0	346.1	4.5	282.0	1187.0	99.8	98.1	0.64	214.3	271.5	57.2	1001.7	148.4
8	Jan. 5	15	62.1	30.53	52.0	0.005715	251.6	407.4	0.0	99.9	100.9	356.7	347.4	6.0	293.6	1187.0	99.8	98.1	0.64	214.8	264.3	49.5	1002.2	147.8
9	Jan. 5	15	63.5	30.52	52.6	0.005737	252.9	407.7	0.0	100.0	101.0	356.7	347.5	6.0	293.6	1187.0	99.8	98.1	0.64	215.1	278.4	63.3	1003.9	147.4
10	Jan. 5	15	65.1	30.48	56.2	0.007405	253.7	407.8	0.0	100.0	101.0	356.7	347.7	6.3	293.4	1187.0	99.8	98.1	0.64	215.0	283.0	68.0	1003.9	147.4
11	Jan. 5	15	67.5	30.46	55.3	0.006387	251.9	407.5	0.0	100.0	101.0	356.6	347.8	6.2	293.1	1187.0	99.8	98.1	0.44	214.3	277.9	63.6	1004.6	148.3
12	Jan. 5	15	66.0	30.48	57.7	0.008014	251.9	407.0	0.0	99.8	100.8	356.3	347.6	6.2	293.1	1187.0	99.0	98.1	0.64	214.7	264.8	50.1	1004.3	147.9
13	Jan. 6	15	55.3	30.43	53.9	0.006096	266.9	413.1	0.0	105.0	105.0	407.4	350.6	6.3	292.4	1186.5	99.6	98.0	0.64	214.6	286.1	71.5	1003.8	156.1

FIG. 8

FIG. 8.—TABLE I.—INTERCO

		Heat In Steam Above Temp. Corresponding To Exhaust Pressure, B.t.u.	Energy Available By Adiabatic Expansion, B.t.u. Per Lb.	Steam Used Per Hour, Lb.	Vacuum At Pump, Inches, Mercury	Vacuum In Intercooler, Inches Of Mercury	Absolute Pressure At Pump, Inches, Mercury	Absolute Pressure In Intercooler, Inches, Mercury	Ratio Of Compression First Stage	Ratio Of Compression Second Stage	Air Temperature At Pump, Degrees, Fahr.	Temp. Of Mixture At Discharge From First Stage, Deg. Fahr.	Temp. Of Mixture At Second Stage Suction, Deg. Fahr.	Water Temperature Entering Intercooler, Degrees Fahr.	Water Temperature Leaving Intercooler, Degrees Fahr.	Temperature Rise Of Water In Intercooler, Deg. Fahr.	Water Through Intercooler, Gallons Per Min.	Work For Isothermal Compression Of Moist Air, Ft. Lb.	Diameter Of Air Nozzle, Inch	Nozzle Area, Square Inch	Moist Air Through Nozzle Per Hour, Lb.	Dry Air To Pump Per Hour, Lb.	Dry Air Under Vacuum
		24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44	4
5	1063.5	1495	857.0	28.02	24.42	0.90	5.50	6.11	5.50	59.1	224.6		58.1	69.8	11.7	46.8	124.94						
	1058.5	1490	857.0	28.82	24.14	1.10	6.78	5.25	5.23	59.5	222.2		57.7	69.0	11.3	46.8	117.84						
5	1061.7	1497.4	857.0	27.93	23.38	2.40	7.05	2.94	4.44	47.8	198.5		60.1	70.9	10.8	46.8	88.52						
5	1061.9	1555.8	857.0	23.24	19.86	7.08	11.26	1.59	2.77	49.4	187.0		59.9	71.8	11.9	46.8	50.85						
5	1062.3	1495.8	857.0	28.83	25.86	0.39	4.54	11.64	7.10	59.9	223.4		59.9			46.7	155.48	0.0	0.0	0.0	0.0	0
5	1064.7	1498.4	857.0	28.39	28.31	8.63	10.91	1.60	2.87	57.5	187.3		58.9			46.8	52.99						
5	1064.7	1498.4	857.0	27.97	23.37	2.24	6.94	3.10	4.51	58.3	202.8		58.4			46.8	92.72						
5	1062.2	1497.8	857.0	29.77	25.18	0.76	5.37	7.07	5.89	56.4	219.6	80.4	58.6	81.4	21.6	25.6	131.19						
5	1063.9	1497.4	857.0	29.37	23.98	1.15	6.59	5.73	4.85	61.2	220.2	86.3	60.0	81.0	21.0	25.6	117.55						
5	1063.1	1497.6	857.0	29.57	24.62	0.91	5.66	6.44	5.42	64.0	224.3	86.2	60.2	81.4	21.2	25.6	126.68						
5	1064.6	1498.3	857.0	28.83	24.33	1.68	6.13	3.76	5.12	63.9	212.0	81.2	60.7	81.1	20.4	25.6	106.66						
5	1064.3	1497.9	857.0	27.38	21.40	3.09	9.08	2.94	3.48	63.5	201.0	87.0	59.3	80.4	21.1	25.6	82.53						
5	1059.8	1501	857.0	26.00	22.00	0.43	5.03	11.69	6.27	69.7	230.6		71.2	92.2	21.0	25.6	155.23	0.0	0.0	0.0	0.0	0

FIG. 8

I.—INTERCONDENSER RADOJET—HANDLING DRY AIR.

	Steam Temperature Before Throttle, Deg. Fahr.	Superheat Before Throttle, Degrees Fahr.	Steam Pressure At Second Stage, Lb. Per Sq. In.-Gage	Steam Pressure At First Stage, Lb. Per Sq. In.-Gage	Steam Temperature At Second Stage, Deg. Fahr.	Steam Temperature At First Stage, Deg. Fahr.	Throttling Calorimeter Pressure, Lb. Per Sq. In.-Gage	Throttling Calorimeter Temperature, Deg. Fahr.	Total Heat In Steam, B.t.u. Per Pound	Steam Quality At Nozzle Conditions, Per Cent	Steam Quality Referred To 264.7 Lb. Per Sq. In., Per Cent	Exhaust Pressure, Lb. Per Sq. In.-Gage	Corresponding Saturated Steam Temp., Deg. Fahr.	Exhaust Temperature By Thermometer, Degrees Fahr.	Superheat At Exhaust, Degrees Fahr.	Heat In Steam Above Temp. Corresponding To Exhaust Pressure, B.t.u.	Energy Available By Adiabatic Expansion, B.t.u. Per Lb.	Steam Used Per Hour, Lb.	Vacuum At Pump, Inches, Mercury	Vacuum In Intercooler, Inches Of Mercury	Absolute Pressure At Pump, Inches, Mercury	Absolute Pressure In Intercooler, Inches, Mercury	Ratio Of Compression First Stage	Ratio Of Compression Second Stage	Air Temperature At
	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	
	393.6	0.0	100.0	99.4	351.9	343.3	5.6	286.8	1184.0	99.5	97.8	0.15	212.5	277.1	64.6	1003.5	149.0	857.0	29.02	24.42	0.90	5.50	6.11	5.50	
	395.4	0.0	100.0	100.0	351.9	344.2	6.0	286.8	1184.0	99.6	97.8	0.15	212.5	277.6	65.1	1003.5	149.0	857.0	28.82	24.14	1.10	5.78	5.25	5.23	
	404.3	0.0	100.0	101.3	354.9	346.4	4.6	286.6	1184.0	99.4	97.8	0.19	214.3	269.7	55.4	1001.7	147.4	857.0	27.93	23.20	2.40	7.05	2.94	4.44	
	404.1	0.0	114.7	114.5	359.9	350.8	4.6	286.6	1184.0	99.2	97.8	0.44	214.1	259.7	45.6	1001.9	155.8	857.0	23.24	18.06	7.08	11.26	1.59	2.77	
	408.0	0.0	99.8	100.0	355.0	346.0	4.3	282.5	1187.0	99.8	98.1	0.10	212.7	256.2	43.5	1006.3	149.8	857.0	29.63	25.66	0.39	4.84	11.64	7.10	
	408.0	0.0	100.0	101.0	355.2	346.2	4.4	292.8	1187.0	99.6	98.1	0.54	214.3	256.6	42.3	1004.7	148.4	857.0	23.39	18.31	6.83	10.91	1.60	2.87	
	407.8	0.0	100.0	101.0	355.0	346.1	4.5	292.0	1187.0	99.6	98.1	0.54	214.3	271.5	57.2	1004.7	148.4	857.0	27.97	23.27	2.24	6.94	3.10	4.51	
	407.4	0.0	99.9	100.9	356.7	347.4	6.0	293.6	1187.0	99.8	98.1	0.54	214.8	264.3	68.5	1004.2	147.8	857.0	28.77	25.16	0.76	5.37	7.07	6.89	
	407.7	0.0	100.0	101.0	356.7	347.5	6.0	293.6	1187.0	99.8	98.1	0.64	215.1	278.4	63.3	1003.9	147.4	857.0	29.37	23.93	1.15	6.59	5.73	4.83	
	407.8	0.0	100.0	101.0	356.7	347.7	6.3	293.4	1187.0	99.8	98.1	0.64	215.0	283.0	68.0	1003.9	147.4	857.0	29.57	24.62	0.91	5.06	6.44	5.42	
	407.5	0.0	100.0	101.0	356.6	347.8	6.2	293.1	1187.0	99.8	98.1	0.44	214.3	277.9	63.6	1004.6	148.3	857.0	28.83	24.33	1.63	6.13	3.76	5.12	
	407.0	0.0	99.8	100.8	356.3	347.8	6.2	293.1	1187.0	99.9	98.1	0.64	214.7	264.8	50.1	1004.3	147.9	857.0	27.39	21.40	3.09	9.08	2.94	3.48	
	412.1	0.0	105.0	105.0	407.4	350.6	6.3	292.4	1186.5	99.6	98.0	0.54	214.6	286.1	71.5	1003.8	150.1	857.4	30.00	25.40	0.43	5.03	11.69	6.27	

FIG. 8

FIG. 8.—TABLE I.—INTERCONDENSER RADOJET—HANDLING DRY AI

Ratio Of Compression First Stage	Ratio Of Compression Second Stage	Air Temperature Air Pump, Degrees Fahr	Temp. Of Mixture At Discharge from First Stage, Deg. Fahr.	Temp. Of Mixture At Second Stage Suction, Deg. Fahr.	Water Temperature Entering Intercooler, Degrees Fahr.	Water Temperature Leaving Intercooler, Degrees Fahr.	Temperature Rise Of Water in Intercooler, Deg. Fahr.	Water Through Intercooler, Gallons Per Min.	Work For Isothermal Compression Of Moist Air, B.t.u.	Diameter Of Air Nozzle, Inch	Nozzle Area, Square Inch	Moist Air Through Nozzle Per Hour, Lb.	Dry Air To Pump Per Hour, Lb.	Dry Air Under Normal Conditions Per Min., Cu. Ft.	Steam Per Pound Of Dry Air, Lb.	Heat Supplied Per Pound Of Dry Air, B.t.u.	Energy Supplied Per Pound Of Dry Air, B.t.u.	Efficiency Based On Heat Supplied, Per Cent	Efficiency Based On Energy Supplied, Per Cent
31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50
6.11	5.50	59.1	224.4		58.1	69.8	11.7	46.8	124.94	[illegible]	[illegible]	54.378	36.348	8.06	23.64	23723	3522	0.629	3.520
5.25	5.23	59.3	222.2		57.7	69.0	11.3	46.8	117.84	[illegible]	[illegible]	58.577	38.364	12.99	14.68	14734	2187	0.803	5.407
2.94	4.44	47.8	198.5		60.1	70.9	10.8	46.8	88.32	[illegible]	[illegible]	135.186	134.631	23.95	6.37	6381	934	1393	9.467
1.59	2.77	49.4	167.0		58.9	71.8	11.9	46.8	50.95	[illegible]	[illegible]	230.462	228.044	31.96	3.60	3607	961	1421	9.151
1.64	7.10	58.9	227.4		58.9			46.7	153.48	0.0	0.0	0.0	0.0	0.0					
1.60	2.87	57.5	182.3		58.9			46.8	52.95	[illegible]	[illegible]	237.325	235.346	32.36	3.64	3651	540	1461	9.912
3.10	4.61	58.3	202.0		58.4			46.8	92.72	[illegible]	[illegible]	133.652	132.890	29.38	6.44	6470	985	1.449	9.751
7.07	5.89	56.4	229.6	90.4	58.6	81.4	21.6	25.6	131.19	[illegible]	[illegible]	55.597	33.448	8.21	36.62	36714	5412	0.359	2.436
6.73	4.83	61.2	220.2	86.3	60.0	81.0	21.0	25.6	117.55	[illegible]	[illegible]	69.428	68.000	13.14	14.50	14557	2137	0.614	5.346
6.44	5.42	64.0	224.3	86.2	60.2	81.4	21.2	25.6	126.68	[illegible]	[illegible]	36.774	36.604	8.15	23.48	23572	3461	0.581	3.008
3.76	5.12	63.9	212.0	91.2	60.7	81.1	20.4	25.6	102.58	[illegible]	[illegible]	94.399	93.709	20.87	9.14	9182	1356	1.157	1.857
2.94	3.48	63.6	201.0	87.0	59.3	80.4	21.1	25.6	82.52	[illegible]	[illegible]	182.641	181.189	40.91	4.73	4760	700	1.761	11.89
11.69	6.27	69.7	230.6		71.2	92.3	21.0	25.6	155.23	0.0	0.0	0.0	0.0	0.0					

LING DRY AIR.

Reference Number																																
Date, 1920																																
Duration Of Run, Minutes																																
Room Temperature, Deg Fahr.																																
Barometric Press. Inches Hg																																
Wet Bulb Temp. Deg Fahr.																																
Water Vapor Per Lb Of Dry Air, Lb																																
Steam Press. Before Throttle Lb. Per Sq. In. Gage																																
Steam Temperature Before Throttle, Deg. Fahr.																																
Superheat Before Throttle Deg. Fahr. (Apparent)																																
Steam Pressure At First Stage, Lb. Per Sq. In. Gage																																
Steam Pressure At Second Stage, Lb. Per Sq. In. Gage																																
Steam Temp. At First Stage, Deg. Fahr.																																
Steam Temp. At Second Stage, Deg. Fahr.																																
Throttling Calorimeter Press. Deg. Fahr.																																
Throttling Calorimeter Temp. Deg. Fahr.																																
Total Heat in Steam, B.t.u. Per Lb.																																
Steam Quality At Nozzle Condition Per Cent Dry																																
Steam Quality Referred To 32°F Lb Per Sq. In. Abs., Per Cent Dry																																
Exhaust Press. Lb. Per Sq. In Gage																																
Corresponding Saturated Steam Temp. Deg. Fahr.																																
Exhaust Temp. By Therm. Deg Fahr.																																
Superheat At Exhaust, Deg Fahr.																																
Heat In Steam Above Temp. Corresponding To Exhaust Press. B.t.u. Per Lb.																																
Energy Available By Adiabatic Expansion B.t.u. Per Lb.																																
Steam Used Per Hr., Lb.																																
Vacuum At Pump, In. Hg.																																
Vacuum in Intercooler, In. Hg.																																
Absolute Pressure At Pump Inches Hg.																																
Absolute Pressure In Intercooler Inches Hg.																																
Ratio Of Compression, First Stage																																
Ratio Of Compression, Sec. Stage																																
Air Temperature At Pump Deg. Fahr.																																
Corresponding Saturated Steam Press. Inches Hg.																																
1	1-6	13	64.3	30.42	54.1	0.006987	53.11	491.8	1.0	104.8	101.8	404.8	399.7	5.0	891.1	1184.3	99.6	98.0	0.44	1143	286.0	71.7	1009.8	1908	899.0	29.01	29.27	1.81	9.19	5.08	6.08	40.1	1.422
2	1-6	15	66.3	30.40	58.7	0.008283	54.7	480.4	0.8	100.0	99.9	374.3	369.3	5.7	885.0	1181.9	99.3	97.6	0.44	1143	276.3	61.5	1000.8	1877	892.0	28.91	29.24	1.71	9.96	5.48	6.48	40.4	1.439
3	1-6	15	71.9	30.41	58.2	0.006994	56.3	492.3	0.1	100.0	100.0	397.2	347.8	5.9	894.1	1187.6	99.9	97.6	0.59	214.7	282.6	67.9	998.8	1846.8	910.0	28.94	29.44	1.92	9.47	5.11	6.24	40.8	1.417
4	1-6	15	72.1	30.48	58.8	0.007164	56.3	492.3	0.1	100.0	100.0	358.0	347.8	6.0	894.3	1187.6	99.8	98.1	0.64	214.9	279.3	64.4	1004.7	1876.8	897.0	28.88	29.29	2.32	6.75	2.81	4.70	48.3	1.431
5	1-7	19	70.0	30.39	59.7	0.008348	57.6	474.6	0.4	100.0	100.0	398.0	348.0	4.6	894.6	1188.6	99.9	98.5	0.59	214.6	284.6	69.4	1008.0	1873.6	897.0	28.96	29.43	2.25	7.55	2.64	4.14	90.1	1.423
6	1-7	19	68.1	30.52	58.5	0.007995	57.2	473.9	0.0	100.0	100.0	398.7	348.0	4.6	894.6	1188.5	99.9	98.5	0.64	214.7	284.8	69.4	1007.9	1871.7	897.0	28.96	29.43	2.26	7.50	2.60	4.10	90.1	1.417
7	1-7	19	64.9	30.50	54.1	0.011756	54.4	471.9	0.0	100.0	100.0	398.9	348.0	4.4	893.0	1187.9	99.9	98.3	0.64	214.7	284.9	69.6	1007.8	1874	897.0	28.91	29.37	2.48	8.93	1.99	3.94	90.0	1.417
8	1-7	15	64.6	30.27	54.8	0.008605	57.2	473.8	0.2	100.0	100.0	398.4	347.1	4.4	893.0	1188.3	99.9	98.2	0.64	214.2	287.1	71.8	1006.1	1878	897.0	28.96	29.19	1.92	9.14	2.66	4.08	100.2	1.438
9	1-7	15	70.0	30.22	58.1	0.007567	56.9	471.0	0.0	100.0	100.0	398.4	347.9	4.4	893.0	1187.7	99.9	98.3	0.59	214.3	286.1	71.4	1008.3	1881.3	897.0	28.89	29.02	2.13	9.24	2.33	3.97	100.0	1.426
10	1-7	15	71.7	30.25	60.1	0.008113	57.3	474.6	0.5	100.0	100.0	398.6	348.1	4.4	893.0	1188.1	99.9	98.3	0.59	214.3	285.1	70.8	1008.2	1879	897.0	28.91	29.27	2.44	9.08	2.37	3.91	100.0	1.426
11	1-7	15	72.3	30.24	60.3	0.008277	57.43	474.3	0.0	100.0	100.0	398.6	348.1	4.6	893.2	1188.1	99.9	98.3	0.64	214.1	284.3	70.1	1008.5	1876	897.0	28.91	29.24	2.78	9.00	1.80	4.29	100.0	1.426
12	1-7	15	74.9	30.24	64.3	0.010159	57.2	473.1	0.0	99.8	99.8	398.9	347.8	4.6	894.0	1188.2	99.9	98.3	0.64	214.1	284.9	69.7	1006.1	1878	897.0	28.86	29.17	2.36	6.07	1.81	3.19	100.0	1.426
13	1-7	15	75.6	30.23	62.3	0.009146	57.3	473.7	0.0	99.9	99.9	398.9	348.0	4.6	893.9	1188.3	99.9	98.3	0.64	214.1	284.8	69.6	1006.1	1877	897.0	28.92	29.25	2.07	8.16	2.01	3.93	100.0	1.426
14	1-7	20	78.4	30.23	64.5	0.008762	57.41	474.3	0.2	100.2	100.2	398.6	348.0	4.8	893.9	1188.9	99.9	98.3	0.69	214.8	284.7	69.9	1007.7	1872	897.0	28.97	29.27	2.42	8.96	1.81	3.93	99.8	1.415
15	1-8	15	70.2	30.12	61.8	0.009757	58.1	476.3	0.2	100.0	100.0	398.9	348.9	4.7	897.6	1189.7	99.9	98.6	0.54	214.1	284.9	70.1	1007.1	1881	897.0	28.92	29.26	0.98	6.48	1.38	4.01	64.0	0.972
16	1-8	19	70.9	30.11	62.0	0.009794	57.6	474.7	0.1	100.0	100.0	398.6	347.9	4.5	897.5	1189.5	99.9	98.5	0.54	214.1	284.5	69.8	1007.1	1879	897.0	28.94	29.27	1.07	7.24	1.71	4.20	64.0	0.978
17	1-8	15	67.8	30.06	60.2	0.010512	56.8	474.2	0.0	100.0	100.0	398.5	348.0	4.5	897.2	1189.6	99.9	98.3	0.64	214.0	284.5	70.1	1008.6	1878	897.0	28.87	29.24	1.81	8.50	1.78	3.99	64.1	0.978
18	1-8	19	68.6	30.00	61.4	0.010514	56.9	471.1	0.0	100.0	100.0	398.3	347.9	4.5	897.3	1189.7	99.9	98.3	0.64	214.0	284.6	69.9	1008.3	1878	897.0	28.92	29.27	2.38	8.66	1.90	3.90	99.9	0.998
19	1-8	15	66.4	30.00	58.1	0.011421	56.14	471.8	0.0	100.0	100.0	398.5	348.0	4.3	897.8	1189.8	99.9	98.3	0.64	214.0	284.7	70.1	1008.1	1877	897.0	28.94	29.27	2.38	8.68	1.82	3.98	64.2	0.976
20	1-9	13	71.4	29.71	62.2	0.010977	57.7	474.9	0.0	100.0	100.0	398.7	348.4	3.6	898.5	1190.1	99.9	98.3	0.64	214.0	284.8	70.2	1007.3	1876	897.0	28.96	29.24	2.41	9.21	1.97	3.74	99.9	0.999
21	1-9	15	72.6	29.68	62.5	0.009868	57.6	474.7	0.1	100.0	100.0	398.6	348.0	3.6	898.5	1188.7	99.9	98.3	0.59	214.6	284.0	70.4	1007.1	1873	897.0	28.90	29.23	2.28	6.98	1.91	4.24	100.1	1.446
22	1-9	15	70.8	29.68	61.1	0.009934	57.1	474.9	0.0	100.0	100.0	398.8	348.1	3.6	898.5	1188.9	99.9	98.3	0.64	214.7	283.6	69.7	1007.1	1874	897.0	28.94	29.27	2.41	9.21	1.93	3.76	100.1	1.446
23	1-9	15	71.2	29.69	62.1	0.009520	57.14	475.4	0.2	100.0	100.0	398.8	348.0	3.5	898.5	1188.8	99.9	98.3	0.64	214.7	284.0	69.9	1007.1	1873	897.0	28.92	29.26	2.31	8.87	1.96	3.87	100.0	1.436
24	1-9	19	71.3	29.64	62.8	0.009043	57.48	475.5	0.0	100.0	100.0	398.7	348.0	3.5	898.5	1189.1	99.9	98.3	0.64	214.7	284.5	69.8	1007.4	1872	897.0	28.93	29.25	2.31	8.61	1.96	4.00	100.0	1.436
25	1-9	19	70.3	29.66	60.3	0.009439	57.62	474.6	0.0	100.0	100.0	398.6	347.8	3.6	898.5	1188.9	99.9	98.3	0.64	214.8	284.1	69.9	1007.1	1873	897.0	28.94	29.27	2.26	8.87	1.93	3.98	99.9	1.434
26	1-9	19	68.8	29.64	60.5	0.009468	57.13	473.7	0.0	100.0	100.0	398.7	348.5	3.5	898.3	1188.3	99.9	98.3	0.59	214.3	283.6	69.1	1006.8	1873	897.0	28.94	29.27	2.41	8.78	1.93	3.98	99.9	1.434
27	1-9	19	67.0	29.66	58.3	0.009841	57.43	474.2	0.2	100.0	100.0	398.8	348.0	3.5	898.5	1188.5	99.9	98.3	0.64	214.7	284.0	69.9	1007.1	1873	897.0	28.94	29.27	2.41	8.80	1.85	4.03	100.1	1.441
28	1-9	19	67.1	29.67	58.3	0.009747	57.3	474.1	0.2	100.0	100.0	398.8	348.0	3.5	898.5	1188.5	99.9	98.3	0.64	214.7	284.0	69.9	1007.1	1873	897.0	28.94	29.27	2.41	8.80	1.85	4.03	100.1	1.441
29	1-9	19	71.4	29.67	62.2	0.009637	57.6	475.2	0.6	100.0	100.0	398.6	348.1	3.5	898.5	1188.9	99.9	98.3	0.64	214.7	284.1	71.4	1007.1	1872	897.0	28.94	29.27	2.38	7.19	1.67	4.01	100.0	1.439
30	1-10	17	68.8	30.34	57.8	0.007750	56.4	481.7	0.2	100.0	100.0	398.6	348.1	4.4	899.0	1189.1	99.9	98.3	0.64	214.7	284.1	69.7	1007.1	1874	897.0	28.94	29.27	2.45	8.86	1.71	3.93	100.0	1.437
31	1-10	15	71.6	30.31	60.2	0.008312	56.14	477.1	0.2	100.0	100.0	398.5	348.0	4.6	897.6	1189.0	99.8	98.2	0.64	214.9	284.7	69.2	1008.0	1873	897.0	28.90	29.28	4.02	6.76	1.68	4.66	110.0	1.587
32	1-10	15	71.3	30.32	59.6	0.007899	56.16	477.2	0.3	100.0	100.0	398.5	348.0	4.6	897.6	1189.0	99.8	98.2	0.64	214.9	284.7	69.2	1008.0	1873	897.0	28.94	29.28	4.78	8.28	1.78	3.79	110.1	1.591
33	1-12	10	64.0	30.18	52.6	0.009639	56.3	469.9	0.3	100.0	100.0	398.7	348.6	4.7	897.6	1187.9	99.9	98.3	0.64	214.3	283.5	69.1	1008.0	1873	897.0	28.94	29.28	1.77	6.66	4.85	4.71	78.1	0.876
34	1-12	15	63.9	30.18	51.5	0.009277	56.5	471.4	0.5	100.0	100.0	398.5	348.5	4.8	897.4	1187.7	99.9	98.3	0.64	214.0	283.5	69.1	1008.0	1873	897.0	28.94	29.28	1.74	6.90	3.89	3.88	78.1	0.876
35	1-12	15	63.9	30.18	51.8	0.009691	56.5	471.4	0.4	100.0	100.0	398.5	348.5	4.8	897.4	1187.7	99.9	98.3	0.64	214.0	283.5	69.1	1008.0	1873	897.0	28.94	29.28	1.74	6.90	3.89	3.88	78.1	0.876
36	1-12	15	63.9	30.18	51.8	0.009691	56.5	471.4	0.4	100.0	100.0	398.5	348.5	4.8	897.4	1187.7	99.9	98.3	0.64	214.0	283.5	69.1	1008.0	1873	897.0	28.94	29.28	1.74	6.90	3.89	3.88	78.1	0.876
37	1-12	15	63.9	30.18	51.8	0.009691	56.5	471.4	0.4	100.0	100.0	398.5	348.5	4.8	897.4	1187.7	99.9	98.3	0.64	214.0	283.5	69.1	1008.0	1873	897.0	28.94	29.28	1.74	6.90	3.89	3.88	78.1	0.876
38	1-12	15	63.9	30.18	51.8	0.009691	56.5	471.4	0.4	100.0	100.0	398.5	348.5	4.8	897.4	1187.7	99.9	98.3	0.64	214.0	283.5	69.1	1008.0	1873	897.0	28.94	29.28	1.74	6.90	3.89	3.88	78.1	0.876
39	1-12	15	63.9	30.18	51.8	0.009691	56.5	471.4	0.4	100.0	100.0	398.5	348.5	4.8	897.4	1187.7	99.9	98.3	0.64	214.0	283.5	69.1	1008.0	1873	897.0	28.94	29.28	1.74	6.90	3.89	3.88	78.1	0.876
40	1-13	15	67.7	29.76	57.6	0.007872	56.4	481.4	0.5	100.0	100.0	398.4	348.5	5.3	897.7	1187.7	99.9	98.3	0.64	213.9	284.7	71.2	1006.3	1868	898.0	28.96	29.28	1.18	4.78	4.10	6.48	78.1	0.876
41	1-13	20	66.8	29.72	56.6	0.007423	56.4	480.8	0.1	100.0	100.0	398.4	347.9	5.0	897.8	1187.0	99.8	98.1	0.64	213.9	285.1	69.6	1008.5	1868	898.0	28.96	29.27	3.96	9.19	2.91	3.97	78.1	0.876
42	1-14	15	78.2	29.84	65.3	0.006177	57.8	482.2	2.4	114.3		398.8		3.6	898.3	1187.1	99.8	98.2	0.64	214.2	283.8	71.1	1009.4	1871	898.0	28.96	29.28	2.68	6.64	1.98	4.41	78.1	0.876
43	1-14	15	78.0	29.90	64.6	0.009491	56.8	484.9	2.3	128.0		398.6		4.0	898.6	1187.8	99.8	98.2	0.64	213.9	283.7	69.9	1008.2	1867	898.0	28.96	29.28	3.81	7.97	1.82	3.98	90.5	1.446
44	1-14	15	78.1	29.91	64.8	0.009360	56.9	485.0	0.5	128.0		398.6		3.8	898.6	1188.0	99.8	98.2	0.64	213.7	284.8	71.1	1008.2	1865	898.0	28.96	29.28	4.34	6.48	1.85	4.26	78.2	0.877
45	1-14	15	78.7	29.91	64.8	0.008411	56.8	483.2	1.1	128.0		398.9		3.9	898.3	1187.6	99.8	98.2	0.64	214.1	284.8	72.7	1008.9	1864	898.0	28.96	29.28	5.15	5.00	0.98	6.24	98.1	1.422
46	1-14	15	78.9	29.87	63.3	0.008467	56.8	482.8	0.4	128.0		398.8		3.8	898.3	1187.6	99.8	98.2	0.64	213.7	283.1	69.4	1008.4	1869	898.0	28.94	29.28	5.53	5.19	0.93	6.31	98.1	1.422
47	1-14	15	78.1	29.87	66.0	0.008083	56.7	482.2	0.4	128.0		398.6		3.9	898.1	1187.9	99.8	98.2	0.64	214.0			1008.4	1869	898.0	28.96	29.28	6.64	6.41	0.97	4.86	98.1	1.422
48	1-14	15	77.9	29.86	63.7	0.009279	56.9	482.1	0.8	128.0		398.9		3.8	898.8	1187.6	99.8	98.2	0.64	214.0			1008.3	1869.2	898.0	28.96	29.28	22.11	22.20	1.78	7.66	8.49	4.87	98.0	1.417
49	1-14	15	78.8	29.86	64.0	0.008854	57.1	484.0	0.9	128.0		398.3		3.8	898.8	1188.2	99.8	98.2	0.64	213.7			1006.9	1869	898.0	28.96	29.28	9.35	9.84	0.99	3.98	98.2	1.420
50	1-15	60	64.9	30.04	56.6	0.005338	57.0	483.2	0.9	100.8	100.0	398.8	348.9	5.3	893.5	1187.9	99.9	98.2			277.9		1010.2	1872	897.0	28.96	29.28	4.77	11.30	1.93		98.5	1.441
51	1-15	60	68.3	29.99	54.3	0.005954	56.9	482.8	0.4	99.8	98.9	398.9	348.7	5.3	893.6	1187.7	99.8	98.2			283.7		1010.2	1872	897.0	28.96	29.28	3.26	9.67	2.80		98.0	1.419
52	1-19	60	64.4	29.94	53.7	0.005753	56.9	482.9	0.2	128.0	128.0	398.8	348.7	5.3	893.6	1187.7	99.8	98.2			284.2		1010.2	1872	897.0	28.96	29.28	2.74	12.26	2.72		98.2	1.424
53	1-19	15	66.0	29.90	53.9	0.009948	57.14	483.7	0.0	100.0	100.0	398.8	348.8	5.3	893.5	1188.0	99.9	98.3	0.64	213.9	284.9	70.0	1009.2	1867	898.0	28.96	29.28	1.73	8.10	1.93	3.85	98.4	1.426
54	1-19	15	65.3	29.88	53.0	0.006092	57.0	483.1	0.2	100.0	100.0	398.8	348.8	5.3	893.7	1187.9	99.9	98.3	0.64	213.9	284.9	70.0	1009.2	1867	898.0	28.96	29.28	1.73	8.10	1.93	3.85	98.4	1.426
55	1-16	29	64.4	29.99	54.0	0.006462	57.19	483.9	0.2	100.0	99.8	398.8	348.8	5.0	893.6	1188.1	99.9	98.3	0.64	213.9	284.8	69.9	1009.2	1867	898.0	28.96	29.28	1.84	9.43	1.97	3.73	98.0	1.417
56	1-16	29	64.0	29.92	53.4	0.006594	56.9	481.3	0.1	100.0	100.0	398.8	348.7	5.0	893.5	1187.9	99.9	98.3	0.64	213.9	284.8	69.9	1009.2	1867	898.0	28.96	29.28	1.80	9.68	1.94	3.68	98.0	1.417
57	1-17	15	63.9	30.04	52.7	0.005772	56.6	481.3	0.4	100.0	100.0	398.8	348.8	4.7	893.8	1187.8	99.9	98.3	0.64	214.9			1009.2	1873	898.0	28.96	29.28	1.41	8.30	1.99	3.70	99.1	1.419
58	1-17	20	64.4	30.02	54.1	0.005905	56.7	481.8	0.1	100.0	100.0	398.6	348.8	4.7	893.6	1187.9	99.9	98.3	0.64	214.9			1009.2	1873	898.0	28.96	29.28	1.41	8.30	1.84	3.85	98.1	1.405

Fig. 9

Fig. 9.—Table II.—Intercondenser Radojet—Hand

31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50	51	52	53	54	55
59	6.06	98.1	1.921			200.1	183	38.8	84.9	150	135.6		Blank.	0.0										
58	5.29	98.6	1.939	0.2779	0.00052	199.0	99.9	98.8	83.2	144	136	178.7	0.196283	0.00737	23.38	23.19	5.19	384.1	98.723	37105	97072	3472	0.936	3.272
11	5.24	98.8	1.917	0.2503	0.00512	207.9	96.8	97.0	81.6	144.6	136	192.9	0.19683	0.00388	24.47	24.32	5.06	394.8	99.486	73266	192659	3413	0.946	4.462
11	4.70	98.5	1.931	0.2898	0.00244	206.0	96.8	98.1	81.1	144.0	136	181.4	0.20544	0.00202	24.64	24.77	15.18	461.6	118.967	14.46	14928	1.34	0.905	5.188
64	4.18	98.1	1.922	1.4218	0.00241	207.8	97.3	97.9	81.4	144.2	136	113.4	0.19757	0.00202	23.70	23.49	28.07	490.2	120.466	922	9279	1.261	1.223	8.336
60	3.95	98.8	1.917	1.143	0.00196	197.16	98.3	98.1	81.1	144.2	136	98.1	0.19777	0.00202	23.74	23.58	33.97	413.7	120.466	6.51	65460	988	1.499	10.245
59	3.94	98.8	1.917	3.065	0.00179	197.0	98.2	98.8	81.7	144.9	136	84.6	0.19802	0.00202	23.65	23.76	34.79	402.7	120.466	4.80	48244	710	1.764	11.965
58	6.08	100.2	1.938			206.0	98.9	98.6	81.6	144.0	136		Blank.	0.0										
33	5.97	100.0	1.934	0.3714	0.00217	204.2	99.2	98.8	83.4	145.2	136	111.7	0.19688	0.00202	23.46	23.47	5.11	447.11	108.64	37.38	37156	3548	0.488	3.687
77	5.41	100.0	1.934	0.3544	0.00195	205.8	98.8	98.1	83.1	145.2	136	103.9	0.19688	0.00202	23.63	23.58	5.11	441.4	108.64	33.89	33718	3528	0.442	4.536
80	6.25	100.0	1.934	0.3854	0.00202	204.0	96.8	97.9	82.6	144.7	136	134.4	0.19644	0.00202	23.38	23.58	5.15	435.1	108.64	14.81	14909	2.99	0.902	6.114
81	5.18	100.0	1.934	1.434	0.00202	198.1	97.7	98.1	81.9	144.5	136	114.8	0.19727	0.00202	23.46	23.49	28.11	404.2	108.64	4.19	4254	1376	1.223	8.326
80	3.95	100.0	1.934	2.144	0.00202	198.8	97.0	98.8	80.9	144.4	136	99.1	0.19644	0.00202	23.26	23.37	39.60	427.7	108.64	6.51	6944	988	1.499	10.221
81	3.83	99.8	1.928	1.048	0.00202	197.0	96.3	99.4	80.3	144.4	136	88.6	0.19552	0.00202	23.30	23.18	39.77	421.6	108.064	4.74	48.7	748	1.177	12.151
80	4.91	64.0	0.998			206.0	91.9	92.7	82.6	149	136		Blank.	0.0										
77	4.90	64.6	0.998	0.577	0.00202	210.8	93.9	92.2	82.1	149	136	143.1	0.19683	0.00202	24.11	23.96	7.96	498.2	92.578	7.67	7107	880	0.584	4.069
78	5.99	64.7	0.998	1.599	0.00194	210.0	94.5	94.8	82.0	149	136	118.9	0.19727	0.00202	23.94	24.17	20.42	498.8	114.988	4.32	4972	1386	1.174	8.177
86	5.90	98.1	1.928	1.989	0.00202	208.0	97.9	98.4	81.9	144.9	136	114.3	0.19552	0.00202	23.74	23.89	28.91	427.8	111.057	6.60	6598	979	1.481	10.318
83	5.48	64.4	0.998	2.880	0.00202	198.5	97.9	98.4	81.6	144.7	136	83.3	0.19552	0.00202	23.74	23.58	39.68	415.0	111.057	4.84	4837	749	1.183	11.499
77	3.74	99.5	1.921			211.1	92.6	93.2	82.7	144.9	136		Blank.	0.0										
81	6.24	100.1	1.940	0.3882	0.00202	214.0	104.3	94.9	83.9	145.4	136	118.9	0.19683	0.00202	23.63	23.46	4.99	488.0	109.656	38.30	38322	4648	0.451	3.804
89	5.76	100.1	1.940	0.592	0.00202	214.0	108.7	97.0	83.8	145.4	136	155.1	0.19683	0.00202	23.94	23.84	7.84	481.7	111.057	28.81	28808	3429	0.633	4.216
86	5.57	100.3	1.938	0.9882	0.00202	214.0	110.3	98.1	83.8	145.2	136	134.4	0.19727	0.00202	23.58	23.77	13.65	474.5	111.057	13.83	13833	1.647	0.987	5.988
85	4.68	100.0	1.938	1.672	0.00202	214.0	110.6	98.8	83.2	144.7	136	116.5	0.19727	0.00202	23.65	23.61	23.85	462.5	111.057	9.68	9680	1.412	1.202	8.621
85	3.85	99.9	1.938	2.231	0.00201	199.8	111.2	99.9	83.8	145.2	136	98.0	0.19727	0.00202	23.54	23.46	33.91	448.0	111.057	6.89	6736	998	1.454	13.005
88	3.18	99.9	1.938	3.341	0.00202	198.0	110.3	99.9	83.6	145.2	136	81.8	0.19802	0.00202	23.54	23.58	38.99	438.8	111.057	4.84	4953	727	1.663	11.954
89	6.43	100.1	1.951	0.3813	0.00178	214.8	108.8	98.8	84.0	145.4	136	174.3	0.19683	0.00202	23.70	23.60	5.03	516.0	111.526	39.92	38.16	4688	0.482	3.663
87	6.01	100.6	1.949	0.441	0.00194	216.1	113.5	99.8	83.8	145.4	136	136.7	0.19683	0.00202	23.61	23.51	7.09	515.6	111.526	28.81	28.43	3430	0.640	4.336
71	5.35	100.0	1.941	0.841	0.00202	214.5	111.9	99.8	83.8	145.4	136	124.7	0.19644	0.00202	23.58	23.51	13.91	499.1	111.057	13.83	13.77	1639	0.975	6.520
78	4.66	100.0	1.941	1.434	0.00202	214.0	109.4	99.1	83.8	145.4	136	110.9	0.19644	0.00202	23.58	23.70	23.43	462.8	111.057	9.22	9.18	1561	1.548	8.890
75	3.79	100.1	1.941	2.137	0.00202	214.0	110.1	98.8	83.2	145.2	136	90.6	0.19644	0.00202	23.58	23.46	34.20	434.6	111.057	6.51	6.23	952	1.914	10.992
80	4.71	73.2	0.878	0.4911	0.00202	211.4	104.5	98.2	83.4	145.2	136	144.6	0.19683	0.00202	23.70	24.11	6.88	492.1	78.695	38.30	37.51	3.486	0.632	4.288
80	5.80	73.5	0.878	0.7611	0.00202	204.9	101.2	98.2	83.6	145.2	136	174.8	0.19644	0.00202	23.70	23.61	13.91	457.0	78.695	28.81	28.74	2.713	0.878	5.956
78	4.93	73.6	0.878	1.041	0.00202	208.0	98.2	98.4	83.8	145.2	136	111.4	0.19683	0.00202	23.87	23.97	20.92	444.3	78.695	13.83	13.78	1955	1.267	8.218
79	4.19	73.4	0.878	1.507	0.00202	198.0	98.8	98.8	83.2	145.2	136	97.0	0.19727	0.00202	23.82	23.61	29.59	427.1	106.302	6.51	6.42	984	1.852	12.111
93	3.96	73.6	0.878	2.103	0.00202	198.0	98.3	98.7	84.1	145.4	136	87.6	0.19802	0.00202	23.82	23.87	38.68	398.8	212.116	4.98	4.96	704	1.722	11.348
10	6.48	73.1	0.878	3.341	0.00202	198.0	100.1	98.8	83.8	145.2	136	127.1	0.19683	0.00202	23.80	23.82	6.05	502.9	63.364	37.81	38.09	3535	0.470	3.468
10	5.57	73.1	0.878	0.8811	0.00202	199.6	98.8	98.8	84.1	145.2	136	161.0	0.19683	0.00202	23.77	23.82	14.50	488.4	205.808	4.85	4.87	722	1.582	11.567
10	5.41	98.5	1.928	0.3901	0.00202	202.1	102.0	98.8	83.8	145.2	136	94.4	0.19683	0.00202	23.70	23.89	5.15	441.2	110.093	38.30	38.41	4117	0.581	4.281
92	5.95	100.1	1.940	0.9101	0.00202	204.0	104.0	98.8	83.8	145.2	136	138	0.19683	0.00202	23.74	23.89	13.91	417.2	111.526	28.81	28.93	4118	0.987	7.286
88	5.24	98.1	1.921	1.434	0.00202	204.0	103.4	98.8	83.8	145.2	136	96.7	0.19683	0.00202	23.70	23.77	20.92	396.6	111.057	13.83	13.79	1.967	1.267	8.218
88	4.82	98.1	1.921	1.903	0.00202	203.0	103.4	98.8	83.8	145.2	136	84.7	0.19683	0.00202	23.70	23.77	29.59	387.1	111.057	9.68	9.67	1378	1.847	11.980
88	4.61	98.1	1.921	2.365	0.00202	203.0	103.4	98.8	83.8	145.2	136	75.1	0.19683	0.00202	23.70	23.77	38.68	378.0	111.057	6.89	6.89	1000	1.726	11.193
88	4.19	98.1	1.921	2.827	0.00202	203.0	103.4	98.8	83.8	145.2	136	67.0	0.19683	0.00202	23.70	23.77	47.46	368.6	111.057	5.17	5.16	749	1.617	10.488
88	3.96	98.1	1.921	3.289	0.00202	203.0	103.4	98.8	83.8	145.2	136	59.7	0.19683	0.00202	23.70	23.77	56.24	359.4	111.057	4.31	4.30	624	1.517	9.840
88	3.83	98.1	1.921	3.751	0.00202	203.0	103.4	98.8	83.8	145.2	136	53.7	0.19683	0.00202	23.70	23.77	65.02	350.1	111.057	3.69	3.68	534	1.427	9.254
88	3.74	98.1	1.921	4.213	0.00202	203.0	103.4	98.8	83.8	145.2	136	48.6	0.19683	0.00202	23.70	23.77	73.80	340.8	111.057	3.23	3.22	467	1.347	8.668
88	3.68	98.1	1.921	4.675	0.00202	203.0	103.4	98.8	83.8	145.2	136	44.4	0.19683	0.00202	23.70	23.77	82.58	331.5	111.057	2.87	2.86	415	1.277	8.082
88	3.64	98.1	1.921	5.137	0.00202	203.0	103.4	98.8	83.8	145.2	136	41.2	0.19683	0.00202	23.70	23.77	91.36	322.2	111.057	2.58	2.57	372	1.217	7.495
88	3.61	98.1	1.921	5.599	0.00202	203.0	103.4	98.8	83.8	145.2	136	38.6	0.19683	0.00202	23.70	23.77	100.1	312.9	111.057	2.35	2.34	339	1.167	6.908
88	3.58	98.1	1.921	6.061	0.00202	203.0	103.4	98.8	83.8	145.2	136	36.5	0.19683	0.00202	23.70	23.77	108.9	303.6	111.057	2.16	2.15	311	1.127	6.321
88	3.56	98.1	1.921	6.523	0.00202	203.0	103.4	98.8	83.8	145.2	136	34.8	0.19683	0.00202	23.70	23.77	117.7	294.3	111.057	2.01	2.00	289	1.096	5.734
88	3.54	98.1	1.921	6.985	0.00202	203.0	103.4	98.8	83.8	145.2	136	33.4	0.19683	0.00202	23.70	23.77	126.5	285.0	111.057	1.89	1.88	271	1.076	5.147
88	3.53	98.1	1.921	7.447	0.00202	203.0	103.4	98.8	83.8	145.2	136	32.2	0.19683	0.00202	23.70	23.77	135.3	275.7	111.057	1.78	1.77	255	1.065	4.560
88	3.52	98.1	1.921	7.909	0.00202	203.0	103.4	98.8	83.8	145.2	136	31.2	0.19683	0.00202	23.70	23.77	144.1	266.4	111.057	1.68	1.67	240	1.055	3.973
88	3.51	98.1	1.921	8.371	0.00202	203.0	103.4	98.8	83.8	145.2	136	30.3	0.19683	0.00202	23.70	23.77	152.9	257.1	111.057	1.59	1.58	227	1.045	3.386
88	3.50	98.1	1.921	8.833	0.00202	203.0	103.4	98.8	83.8	145.2	136	29.6	0.19683	0.00202	23.70	23.77	161.7	247.8	111.057	1.51	1.50	215	1.037	2.799
88	3.49	98.1	1.921	9.295	0.00202	203.0	103.4	98.8	83.8	145.2	136	28.9	0.19683	0.00202	23.70	23.77	170.5	238.5	111.057	1.44	1.43	204	1.030	2.212

Several Things Only In Operation.

Effect Of Increasing T Each Pressure.

Effect Of Excess Moisture.

Diagram In Thermodynamics

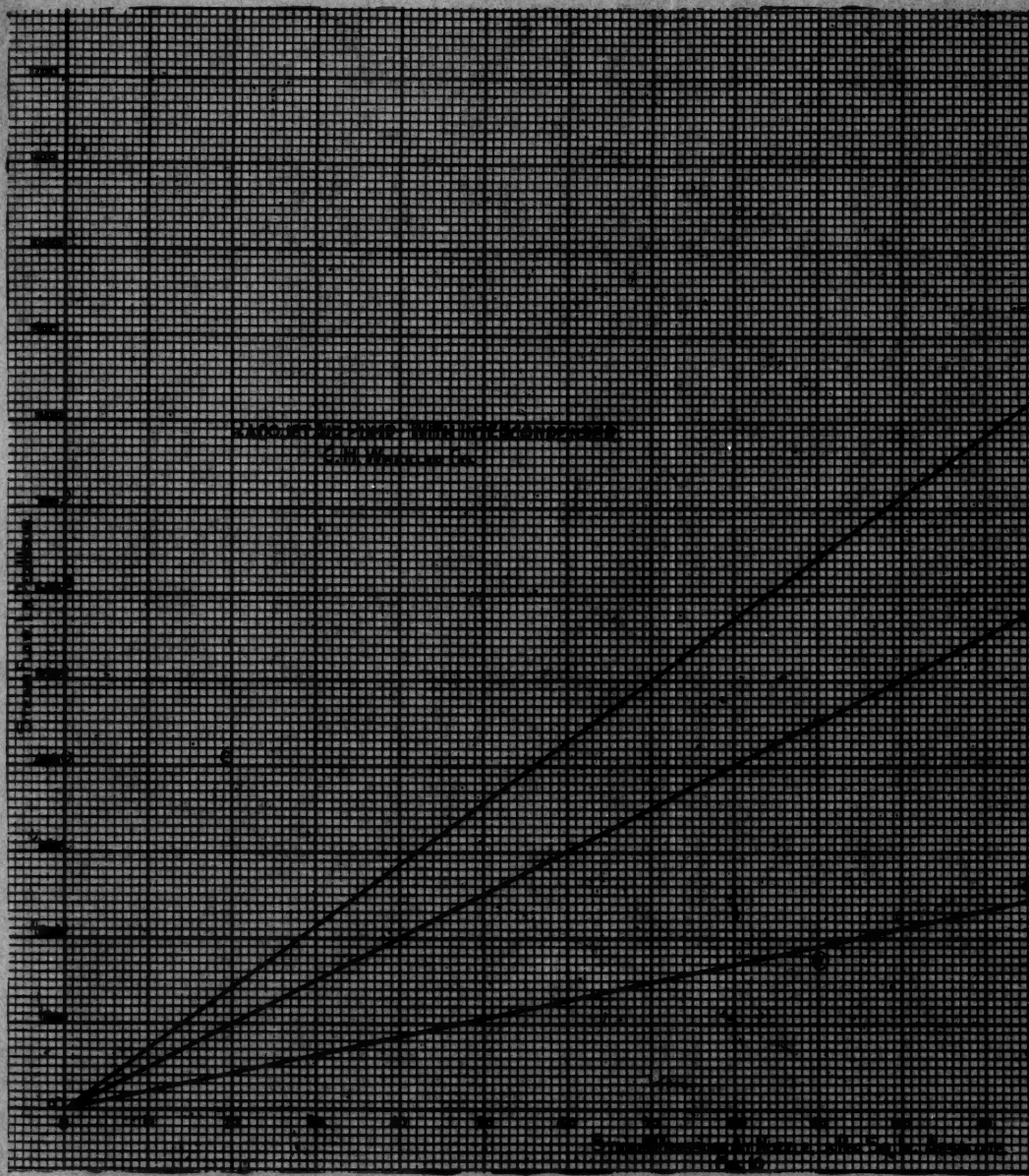
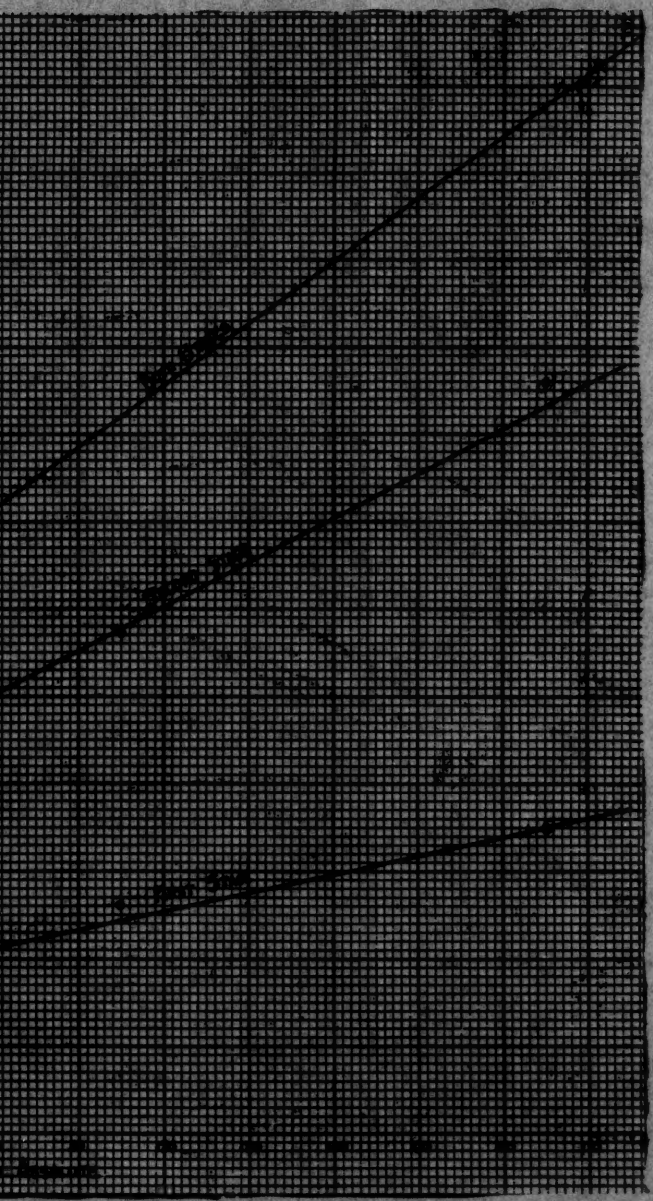


FIG. 10.—STEAM FLOW CURVES—INTERCONDENSER RADOJET.



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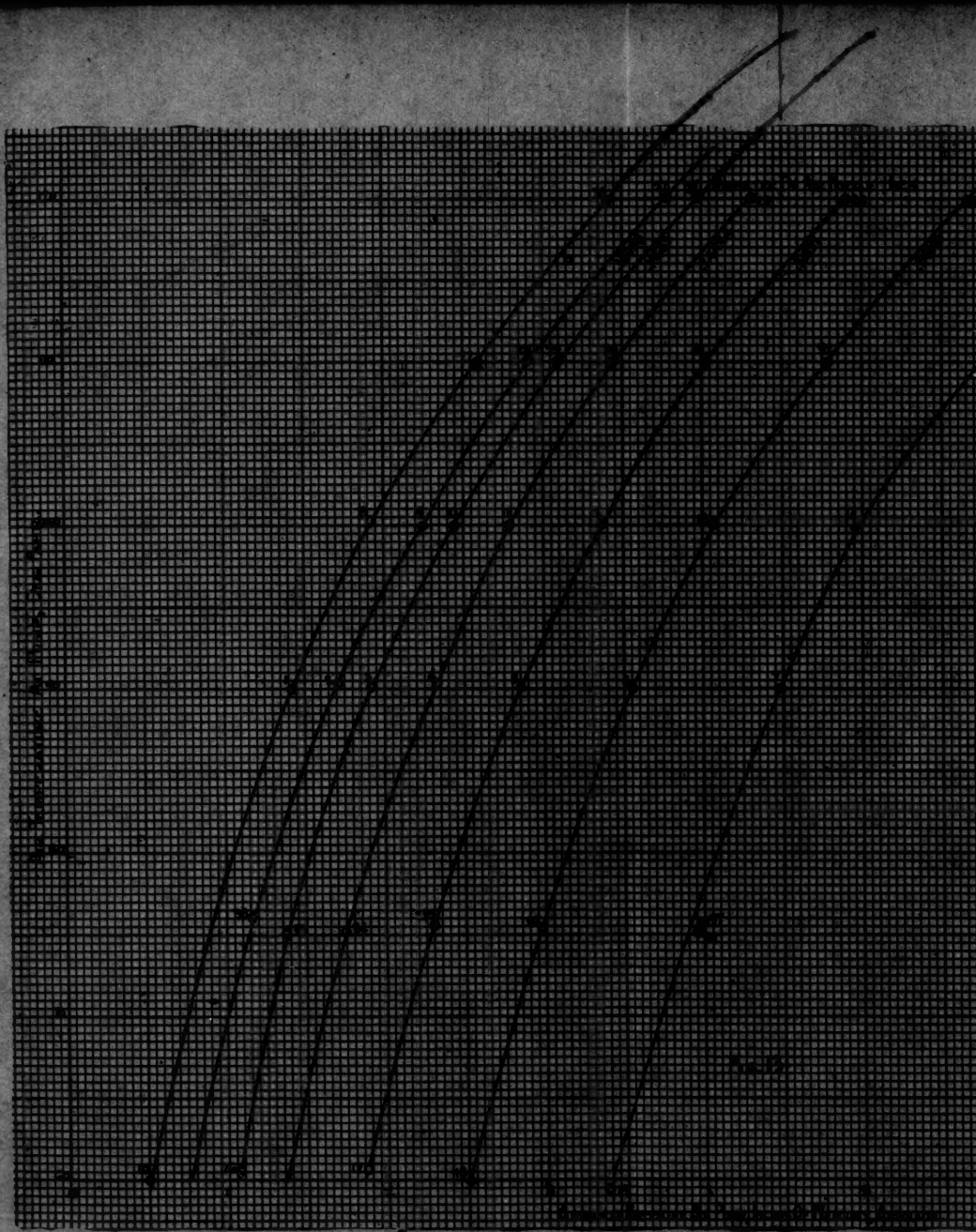
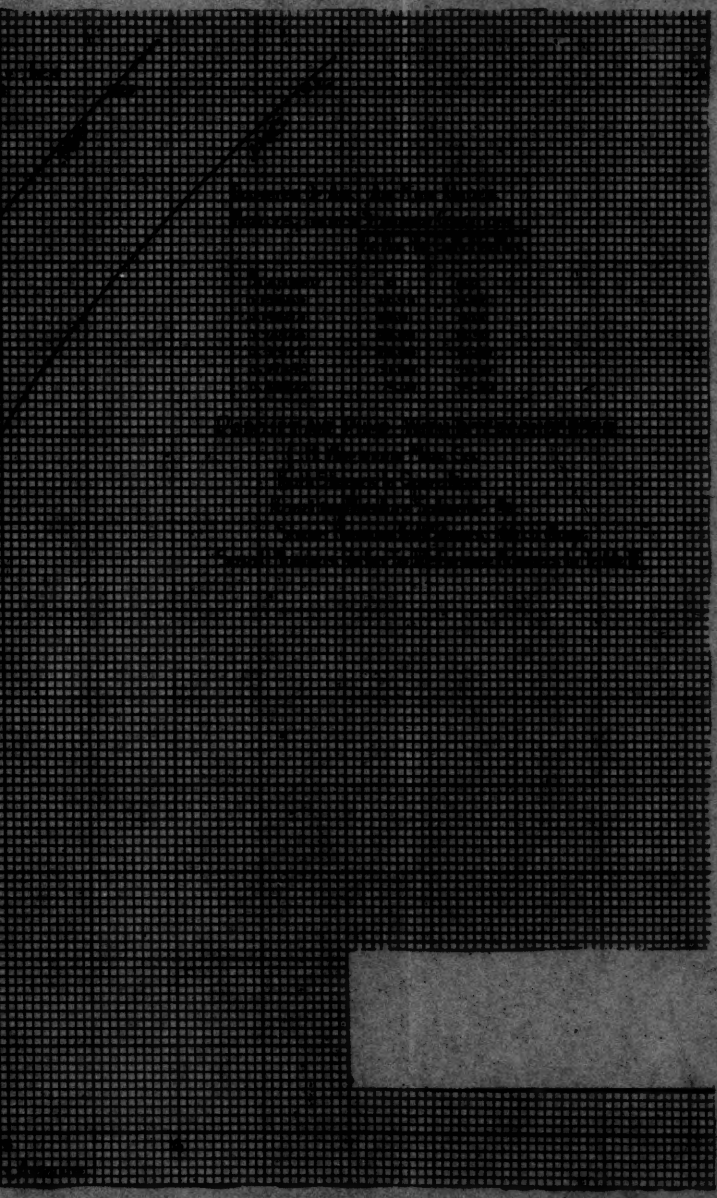


FIG. 13.—CHARACTERISTICS OF INTERCONDENSER RADIOJET—HANDLING M



-Handling Moisture Saturated Air.

Reference.	Date - 1920	Position of Nozzle.	Room Temperature Deg.Fahr.	Barometric Pressure Inches Mercury	Wet Bulb Temperature deg.Fahr.	Water Vapor per pound of Dry Air, lb.	Steam Press. at Second Stage,lb.per sq.in.Gage.	Steam Press.at first stage, lb.per sq.in.Gage.	Steam Temperature at second stage,deg.Fahr.	Steam Temperature at First Stage, Deg.Fahr.	Throttling calorimeter Press.,lb.per sq.in.Gage.	Throttling calorimeter Temperature,Deg.Fahr.	Total Heat in Steam B.T.U. per lb.	Steam Quality at Nozzle Condition, per cent dry.	Steam Quantity referred to 24.7 pounds per sq.in. per hour dry.	Exhaust Pressure lb.per sq.in.gage.	Corresponding saturated Steam Temp. deg.Fahr.	Exhaust Temperature by Thermometer. deg.Fahr.	Temperature at Exhaust deg.Fahr.	Heat in Steam above Temp. corresponding to Exhaust
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21
1	2-17	15	67.0	29.85	53.5	.005628	100.1	98.9	341.0	341.0	0.93	313.0	1197.9	1*	99.5	0.27	212.8	283.5	72.7	101
2	2-17	15	68.7	29.83	54.5	.005846	100.1	98.9	340.3	340.3	0.48	316.6	1199.9	1*	99.8	0.27	212.7	284.5	71.8	101
3	2-17	15	70.1	29.81	56.7	.006754	100.1	98.9	340.4	340.4	0.93	317.2	1200.1	2*	99.8	0.27	212.7	282.9	70.2	101
4	2-17	15	69.9	29.81	55.4	.006082	100.1	98.9	340.5	340.5	0.55	317.0	1200.0	2*	99.8	0.81	214.5	280.9	66.4	101
5	3-1	15	63.6	29.77	52.8	.006056	100.1	98.9	340.1	340.1	0.26	291.0	1187.8	99.9	98.2	1.12	215.4	277.1	61.7	100
6	3-1	15	65.0	29.75	53.1	.006914	100.1	98.9	340.9	340.9	0.26	292.5	1188.0	99.9	98.2	0.24	212.5	283.8	73.3	100
7	3-1	15	68.7	29.74	56.2	.006789	105.1	103.9	343.0	343.0	0.30	296.7	1190.5	2*	98.5	1.12	215.4	278.9	63.5	100

Fig. 2

*Degrees Superheat

FIG. 20.—TABLE I.—RADOJET WIT

Thermodyn. Eqs. (Btu/lb. deg.abs.)	Specific Heat at Const. Vol. (Btu/lb. deg.abs.)	Heat in Fluid above Temp. Corresponding to Exhaust Press. B.t.u. per lb.	Energy available for adiabatic expansion. B.t.u. per hour.	Steam used per hour lb.	Vacuum at Pump Inches Mercury	Pressure at Second Stage Suction. In. of Hg.	Absolute Press at Pump "Hg.	Absolute Pressure at Second stage suction "Hg.	Ratio of Compression 1st stage.	Ratio of Compression 2nd stage.	Air Temp. at Pump Deg. Fahr.	Work for Isothermal Compression of Moist Air. ft.lb. per lb.	Diameter of Air Nozzle Inches.	Nozzle Area Sq.in.	Moist Air through Nozzle per Hr. lb.	Air to Pump Ft per Lb.	Dry Air under normal Conditions, per min. cu.ft.	Steam Per pound of
19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37
73.5	72.7	1017.1	152.47	1720.0	29.19	24.51	0.66	5.92	8.36	5.51	68.2	138.48	0.15683	.019317	22.91	22.78	5.07	75.5
74.5	71.8	1019.2	152.61	—	29.03	24.13	0.80	5.70	7.13	5.33	67.7	131.97	0.14633	.030335	35.88	35.67	7.94	48.1
72.9	70.2	1019.4	152.81	—	28.78	23.93	1.03	5.87	5.70	5.17	67.7	122.24	0.24944	.048868	57.79	57.40	12.77	29.9
70.9	66.4	1017.5	150.41	—	28.19	23.22	1.62	6.39	4.07	4.61	67.7	109.74	0.31527	.078065	92.15	91.59	20.38	18.7
67.1	61.7	1014.3	146.36	—	27.32	21.31	2.85	8.46	3.76	3.98	52.7	91.63	0.37490	.120588	230.93	230.14	28.95	19.2
75.8	73.3	1007.5	150.56	—	29.39	23.79	0.26	5.96	3.73	5.08	60.3	167.04	Diameter	0.0	0.0	0.0	0.0	—
68.9	69.5	1007.0	149.73	—	26.72	20.34	3.02	9.40	3.11	3.22	61.9	82.16	0.43822	.150886	177.82	176.62	39.29	9.7

Fig. 20

LADOJET WITHOUT INTERCONDENSER—HANDLING DRY AIR.

Steam Press. at Second Stage, lb. per sq. in. Abs.	Steam Press. at First stage, lb. per sq. in. Abs.	Steam Temperature at second stage, deg. Fahr.	Steam Temperature at First Stage, Deg. Fahr.	Throttling Calorimeter Press., lb. per sq. in. Abs.	Throttling Calorimeter Temperature, Deg. Fahr.	Total Heat in Steam B.t.u. per lb.	Steam Quality at Nozzle Conditions, per cent dry.	Steam Quality referred to 284.7 pounds per sq. in. per cent dry.	Exhaust Pressure Lb. per sq. in. gage.	Corresponding saturated Steam Temp. deg. Fahr.	Exhaust Temperature by Thermometer, deg. Fahr.	Superheat at Exhaust deg. Fahr.	Heat in Steam above temp. Corresponding to Exhaust Press. B.t.u. per lb.	Energy available by Adiabatic expansion. B.t.u. per hour.	Steam used per hour Lb.	Vacuum at Pump Inches Mercury	Steam at Second Stage Nozzles. Lb. of Sq.	Absolute Press at Pump " Sq
8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26
100.1	98.9	341.0	341.0	0.35	313.0	1197.9	17°	99.5	0.27	212.8	285.5	72.7	1017.1	152.47	1720.0	29.19	24.51	0.66
100.1	98.9	340.3	340.3	0.45	316.6	1199.9	19°	99.8	0.27	212.7	284.5	71.8	1019.2	152.61	--	29.03	24.13	0.80
100.1	98.9	340.4	340.4	0.35	317.2	1200.1	20°	99.8	0.27	212.7	282.9	70.2	1019.4	152.81	--	28.78	23.93	1.03
100.1	98.9	340.5	340.5	0.55	317.0	1200.0	20°	99.8	0.81	214.5	280.9	66.4	1017.5	150.41	--	28.19	23.22	1.62
100.1	98.9	340.1	340.1	0.26	291.0	1187.8	99.9	98.2	1.12	219.4	277.1	61.7	1004.3	146.36	--	27.32	21.31	2.23
100.1	98.9	340.9	340.9	0.26	292.5	1188.0	99.9	98.2	0.24	212.5	283.8	73.3	1007.5	150.36	--	29.39	23.79	0.16
103.1	103.9	343.0	343.0	0.30	296.7	1190.5	2°	98.5	1.12	219.4	278.9	69.5	1007.0	149.73	--	26.72	20.34	3.02

Fig. 20

FIG. 20.—TABLE I.—RADOJET WITHOUT INTERCONDENSER—HANDLING

Efficiency	Volume at Second Stage Section. In. of Hg.	Absolute Press at Fan "Hg	Absolute Pressure at Second stage section "Hg.	Ratio of Compression 1st stage.	Ratio of Compression 2nd stage.	Air Temp. at Fan Deg. Fahr.	Work for Isothermal Compression of Unit Air, B.t.u. per lb.	Diameter of Air Nozzle Inches.	Nozzle Area Sq.in.	Moist Air through Nozzle per Hr. lb.	Dry Air to Fan per hr. lb.	Dry Air under normal conditions, per min. cu.ft.	Steam Per pound of dry air. lb.	Heat supplied per Pound of Dry Air. B.t.u.	Energy Supplied per pound of dry Air. B.t.u.	Efficiency Based on Heat Supplied, Per cent	Efficiency Based on Energy Supplied. Per cent.
25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41
24.31	0.66	5.52	8.36	5.51	68.2	138.46	0.15683	.019317	22.91	22.76	5.07	73.30	76791.1	11521.5	.181	1.21
24.19	0.80	5.70	7.13	5.33	67.7	131.97	0.14653	.030335	35.88	35.67	7.94	48.22	49145.8	7368.9	.269	1.80
23.93	1.03	5.87	5.70	5.17	67.7	122.24	0.24344	.048868	57.79	57.40	12.77	29.97	30552.4	4579.7	.403	2.69
23.22	1.62	6.39	4.07	4.61	67.7	103.74	0.31367	.078065	92.15	91.59	20.38	18.78	19108.7	2824.7	.327	3.77
22.31	2.25	8.46	3.76	3.98	72.7	91.89	0.37490	.110388	130.93	130.14	28.95	13.22	13276.8	1934.9	.690	4.74
23.79	0.16	5.96	3.73	5.08	60.9	167.04	Diameter	0.0	0.0	0.0	0.0	---	0.0	0.0	0.0	0.0
20.54	3.02	9.40	3.11	3.22	61.9	82.16	0.43822	.150886	177.82	176.62	39.29	9.74	9806.2	1458.4	.843	5.67

—Handling Dry Air.

Reference	Run, 1950	Duration of Run, Minutes	Air Temperature, Dry Bulb, °F.	Air Temperature, Wet Bulb, °F.	Air Temperature, Dew Point, °F.	Water Vapor Per Lb. of Dry Air, Grains	Air Pressure Before Nozzle, Lbs. per Sq. In. Abs.	Pressure of Steam Before Nozzle, Lbs. per Sq. In. Abs.	Steam Temperature Before Nozzle, °F.	Air Temperature at Nozzle Throat, °F.	Air Velocity at Nozzle Throat, Ft. per Min.	Air Flow Through Nozzle, Lbs. per Min.	Steam Flow Through Nozzle, Lbs. per Min.	Steam Quality Before Nozzle, %	Steam Quality at Nozzle Throat, %	Air Pressure at Nozzle Throat, Lbs. per Sq. In. Abs.	Air Pressure Before Diffuser, Lbs. per Sq. In. Abs.	Air Temperature Before Diffuser, °F.	Air Pressure After Diffuser, Lbs. per Sq. In. Abs.	Air Temperature After Diffuser, °F.	Air Velocity at Diffuser Discharge, Ft. per Min.	Air Flow Through Diffuser, Lbs. per Min.	Air Pressure Rise Through Diffuser, Lbs. per Sq. In.	Air Temperature Rise Through Diffuser, °F.	Diffuser Efficiency, %
1	2-18	15	71.5	49.81	40.1	0.00848	267.0	100.1	98.9	388.5	388.5	0.4	388.9	1196.2	-	99.3	0.37	222.6	284.0	71.4	1085.7	272.8	17
2	2-18	15	66.1	49.82	34.8	0.00649	265.3	100.1	98.9	398.2	342.2	0.4	383.0	1198.5	-	99.6	0.07	222.2	283.4	72.8	1018.0	291.4	17
3	3-3	15	63.2	50.49	36.6	0.00805	258.5	100.1	98.9	342.1	342.1	0.4	344.2	1176.1	98.5	96.8	0.82	213.7	284.4	70.9	994.3	146.9	17
4	2-19	15	59.8	49.44	49.3	0.00512	257.0	100.1	98.9	339.5	339.5	0.4	336.4	1159.0	-	98.5	0.57	212.6	273.4	62.8	1029.5	131.0	17
5	3-3	10	68.7	50.40	35.2	0.00602	267.7	100.1	98.9	340.3	340.3	0.7	331.7	1187.3	99.8	98.2	0.54	214.5	281.2	66.7	1004.8	148.5	17
6	3-3	15	63.4	48.93	38.9	0.00972	263.5	100.6	100.9	346.0	346.0	0.7	332.3	1178.8	96.8	97.2	1.22	215.4	281.3	65.9	995.4	147.6	17
7	3-3	15	63.7	49.32	38.0	0.00969	262.3	100.6	100.9	343.8	343.8	0.7	333.8	1180.5	96.9	97.3	1.12	215.0	278.5	63.5	997.4	148.0	18
8	2-18	15	73.0	49.75	37.7	0.00649	262.5	100.1	98.9	341.3	341.3	0.5	345.1	1199.0	-	99.7	0.27	222.6	281.8	59.2	1028.5	131.9	17
9	3-2	15	73.5	50.25	37.8	0.00647	266.8	100.1	98.9	341.0	341.0	0.6	335.8	1189.7	-	98.5	0.84	213.5	282.0	68.5	1026.2	149.2	17
10	3-2	15	73.8	50.26	38.7	0.00650	257.0	100.1	98.9	342.1	342.1	0.6	335.8	1189.7	-	98.5	0.84	213.5	281.6	68.1	1026.2	149.2	17
11	3-2	15	73.4	50.26	38.5	0.00655	258.0	100.1	98.9	340.4	340.4	0.6	334.6	1188.9	-	98.3	0.84	213.5	280.1	66.6	1027.4	149.3	17
12	3-2	15	74.2	50.25	38.5	0.00657	264.3	100.1	98.9	341.3	341.3	0.7	335.1	1189.1	-	98.4	1.03	216.5	- 273.6	63.6	1024.9	145.0	17
13	3-3	15	63.6	49.48	39.3	0.00993	273.8	100.6	100.9	348.6	343.6	0.7	286.6	1183.1	93.6	97.9	0.49	213.4	278.6	64.2	1028.5	151.7	17
14	3-3	15	65.6	49.49	41.5	0.01085	269.3	110.6	109.9	347.8	347.8	0.7	280.9	1183.1	93.2	97.4	0.55	*214.4	278.6	64.2	1028.6	159.0	17
15	2-19	15	67.4	49.44	34.3	0.00647	257.0	100.1	98.9	339.9	339.9	0.6	332.6	1176.9	+	99.9	0.27	222.6	277.1	64.5	1026.4	131.8	17
16	3-2	15	74.9	50.22	39.2	0.00793	262.5	100.1	98.9	339.7	339.7	0.8	335.5	1189.6	-	98.4	0.24	213.1	281.8	68.7	1028.3	149.7	17
17	3-2	15	75.3	50.20	38.8	0.00803	261.3	100.1	98.9	340.4	340.4	0.7	334.6	1188.8	+	98.3	0.24	213.1	280.3	67.2	1027.7	149.7	17
18	3-2	15	74.8	50.22	37.1	0.00796	268.0	100.1	98.9	339.9	339.9	0.8	334.5	1189.1	-	98.4	0.24	213.1	279.4	66.3	1028.1	149.5	17
19	3-4	15	69.5	50.24	40.0	0.00871	268.5	103.6	103.9	343.2	343.2	0.5	340.6	1188.2	97.8	98.2	1.32	216.7	278.2	61.5	1029.4	146.5	17
20	3-4	15	73.5	50.26	45.3	0.01080	269.8	110.6	108.9	346.0	346.0	0.6	343.2	1188.8	99.8	98.3	1.48	217.0	278.4	61.4	1029.8	150.6	17
21	3-4	15	74.1	50.28	44.2	0.01120	270.0	115.7	113.9	349.4	343.2	0.5	331.7	1187.8	99.6	98.2	1.48	218.9	278.5	60.0	1028.8	150.0	17
22	3-4a	15	74.6	50.36	40.5	0.00780	261.5	100.1	98.9	339.1	339.7	0.4	331.8	1187.2	-	99.3	0.27	213.5	277.2	63.7	1028.9	150.7	17
23	3-4a	15	74.9	50.28	40.6	0.00778	263.4	100.1	98.9	340.9	340.9	0.4	331.0	1186.6	-	99.3	0.27	213.5	277.6	64.1	1028.2	150.3	17
24	3-4	15	68.8	50.25	44.0	0.00995	273.8	110.6	99.9	343.1	343.1	0.6	331.5	1187.5	98.8	98.2	0.82	218.0	278.6	64.5	1026.8	149.9	17
25	3-4	15	74.6	50.26	44.9	0.00979	273.5	110.6	99.9	340.5	343.9	0.6	331.7	1187.5	98.8	98.2	0.82	218.0	278.1	64.1	1026.8	149.9	17
26	3-5	15	65.2	50.32	39.7	0.00967	278.5	110.6	109.9	347.6	347.8	0.6	283.6	1186.5	93.6	98.1	0.54	218.0	278.6	64.7	1029.6	150.7	17
27	3-5	15	70.7	49.39	43.6	0.01110	260.0	110.6	109.9	346.7	346.7	0.7	286.7	1185.5	93.8	97.9	0.46	218.6	277.2	64.6	1024.9	151.8	17
28	3-5	15	70.7	49.39	47.0	0.01285	267.3	115.7	114.9	349.0	349.0	0.6	283.2	1184.7	93.8	97.8	0.44	222.6	276.9	64.5	1022.1	158.0	17
29	3-5a	15	70.7	50.49	48.3	0.01295	256.3	100.1	98.9	340.1	340.1	0.4	327.6	1175.0	-	99.1	0.82	213.6	273.9	60.3	1013.4	140.8	17
30	3-5	15	70.5	49.39	39.5	0.00896	268.5	105.6	104.9	348.8	343.8	0.6	283.8	1184.0	93.4	97.7	0.44	222.6	276.2	63.6	1028.4	152.2	17
31	3-6	15	73.6	49.80	45.5	0.00918	271.5	110.6	99.9	349.8	343.2	0.6	289.1	1177.3	98.7	96.9	0.28	222.6	278.6	64.0	956.7	148.2	17
32	3-6	15	57.0	49.81	46.2	0.00930	263.0	110.6	99.9	343.8	343.8	0.2	278.0	1176.3	98.8	97.1	0.24	222.6	278.0	64.4	956.2	148.4	17
33	3-6	15	70.3	49.82	47.9	0.00947	270.8	110.6	99.9	346.8	346.8	0.2	273.4	1180.6	99.1	97.3	1.67	227.4	278.4	58.0	956.2	148.4	17
34	3-6	15	70.4	49.82	48.2	0.00956	274.8	115.7	114.9	349.2	349.2	0.2	277.7	1181.0	98.8	97.4	1.58	226.4	278.4	60.0	956.6	139.1	17
35	3-6	15	67.8	49.82	50.2	0.00311	254.3	115.7	114.9	348.5	348.5	0.2	277.9	1181.0	98.8	97.4	1.28	225.2	278.8	59.6	957.7	134.6	17
36	2-26	15	60.4	49.76	49.9	0.00995	270.8	100.1	-	339.5	-	0.6	302.9	1192.3	-	98.7	0.22	228.5	283.1	54.6	1033.8	131.4	17
37	2-26	15	68.4	49.76	51.8	0.01032	276.3	100.1	-	339.4	-	0.6	303.4	1192.1	+	98.9	0.22	228.5	283.4	54.9	1032.7	131.6	17
38	2-26	15	60.7	49.77	50.4	0.00992	274.5	100.1	-	339.2	-	0.6	301.2	1193.9	-	98.9	0.22	232.5	283.4	45.9	1028.9	131.1	17
39	2-26	15	58.3	49.77	48.7	0.00975	278.3	105.1	-	340.2	-	0.6	301.0	1191.9	-	98.6	0.69	234.0	276.7	42.7	1029.8	131.6	17
40	2-26	15	59.6	49.76	49.0	0.00984	275.8	105.1	-	340.0	-	0.5	300.4	1191.9	-	98.6	0.44	233.8	279.2	45.4	1030.2	131.9	17
41	2-26	15	61.1	49.78	51.4	0.01039	268.3	105.1	-	340.2	-	0.5	300.7	1191.8	-	98.6	0.44	233.7	278.4	44.7	1030.2	132.0	17
42	3-7	15	69.7	50.40	54.4	0.01097	264.0	100.6	99.9	340.7	343.7	1.0	290.0	1185.8	99.1	98.2	0.49	234.1	284.1	49.0	1041.7	150.5	17
43	3-7	15	70.9	50.38	54.3	0.01094	265.8	100.6	99.8	343.7	343.7	1.0	291.0	1186.0	99.0	98.2	0.49	234.1	284.8	44.7	1042.9	149.9	17

Fig. 21

FIG. 21.—TABLE II.—(a) RADOJET WITHOUT INTERCONDENSER—HANDLING MOISTURE SATURATED AI

24	25	26	27	28	29	30
1025.7	251.8	1700	29.21	24.21	2.60	5.39
1018.0	251.4	1700	29.05	24.35	2.77	5.47
994.3	146.9	1700	29.39	28.35	2.26	7.13
1009.5	251.0	1700	28.13	23.09	2.91	6.05
1004.8	148.5	1700	28.40	22.11	2.66	5.59
995.4	147.6	1700	28.09	21.16	2.44	5.07
997.4	148.0	1700	28.10	20.36	2.26	4.75
1018.5	251.9	1700	28.33	23.29	2.80	5.85
1008.2	149.2	1700	28.27	22.29	2.54	5.35
1008.2	149.2	1700	28.27	22.27	2.48	7.05
1007.4	148.9	1700	28.29	22.14	2.46	5.11
1004.9	148.8	1700	27.99	21.87	2.44	5.05
1003.7	251.7	1700	28.28	20.35	2.80	5.85
1000.5	253.0	1697	28.44	19.47	4.83	9.85
1026.4	251.8	1700	28.20	23.01	2.66	5.63
1008.3	149.7	1700	28.44	22.32	1.78	7.39
1007.7	149.7	1700	28.43	22.08	2.53	5.33
1006.1	149.5	1700	27.73	21.69	2.48	5.21
1003.4	146.5	1700	27.39	21.35	3.01	6.78
1003.8	254.4	1697	28.48	20.23	3.64	7.75
1002.8	253.0	1701	28.38	19.48	4.83	10.09
1018.9	251.7	1700	27.44	23.05	2.64	7.24
1005.1	149.9	1700	27.47	23.26	2.88	7.48
1006.5	149.9	1700	28.09	22.68	2.46	5.48
1006.1	149.9	1700	28.46	22.57	3.39	8.88
1005.6	253.7	1697	28.47	20.48	3.92	9.73
1002.9	253.8	1701	28.38	19.49	4.85	10.09
1005.1	253.8	1701	28.46	18.55	5.91	11.26
1013.6	149.9	1700	27.46	23.07	3.45	7.38
1005.4	251.8	1700	28.22	22.22	3.37	9.18
994.7	148.2	1700	28.09	22.44	3.91	9.26
998.2	148.4	1700	28.38	22.48	4.51	9.84
995.2	148.8	1700	28.36	22.48	4.56	10.34
994.5	253.1	1701	28.38	20.12	5.27	11.48
991.7	254.6	1701	28.47	18.09	7.15	13.73
1005.8	251.4	1700	27.89	22.27	3.38	7.61
1002.7	251.6	1700	28.19	22.22	3.26	7.38
1000.8	251.1	1700	28.31	21.58	3.26	7.39
1007.8	251.6	1701	28.37	22.26	4.09	9.61
1000.1	252.9	1701	28.38	21.48	4.83	9.88
1000.1	252.9	1701	28.37	21.48	4.83	9.88
1004.7	149.5	1700	28.49	22.24	1.82	7.05
1005.8	149.9	1700	28.49	22.28	1.82	7.05

ATURATED AIR.

	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46	47
1	8.89	9.73	65.9	0.693	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
2	7.10	5.55	65.6	0.695	0.135	.000381	193.7	0.11839	.000721	11.76	11.69	2.60	571.83	45.796	147.76	149341	22259.2	0.151	0.869
3	6.15	4.35	65.1	0.694	0.556	.001555	144.1	0.19655	.030335	36.85	36.64	8.12	1618.08	66.366	47.87	46801	6856.3	0.329	2.101
4	4.94	4.61	65.1	0.694	0.886	.002479	189.6	0.24944	.048868	57.93	57.69	12.85	2563.21	82.855	25.83	30134	4947.4	0.417	2.767
5	4.15	3.80	65.9	0.692	1.238	.003468	111.3	0.31387	.078065	94.10	93.54	20.81	4154.38	180.994	16.59	18098	2730.9	0.602	4.076
6	3.17	3.80	65.2	0.686	2.014	.005640	92.3	0.39830	.110388	149.74	148.48	33.60	6710.63	199.389	13.97	17286	2268.0	0.693	4.681
7	2.68	3.48	65.3	0.689	2.581	.007186	84.3	0.43822	.139886	177.14	175.66	39.49	7884.17	200.009	12.83	16889	1984.3	0.805	5.350
8	4.85	5.12	64.7	1.196	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
9	3.49	4.16	64.9	1.203	0.138	.000368	200.7	0.11839	.009721	11.84	11.78	2.62	527.96	77.164	146.76	147209	21021.9	0.158	0.970
10	2.48	4.08	64.8	1.201	0.478	.001267	153.3	0.19693	.030335	36.18	35.93	7.99	1613.78	91.993	47.87	48063	7168.8	0.328	2.146
11	3.54	3.73	64.9	1.205	0.855	.002282	131.7	0.24944	.048868	58.35	57.96	12.89	2603.08	108.880	29.67	29890	4459.7	0.411	2.673
12	3.26	3.73	64.9	1.205	1.493	.003544	111.8	0.31387	.078065	93.13	92.52	20.58	4135.10	140.814	18.58	18581	2784.8	0.596	4.159
13	2.78	3.44	65.0	1.209	1.991	.004808	99.1	0.37890	.110388	149.51	148.24	33.32	6660.80	176.702	14.00	14892	2123.8	0.795	4.666
14	2.42	3.20	64.8	1.201	2.849	.006940	89.7	0.43822	.139886	176.69	174.75	38.87	4153.60	220.979	10.89	10916	1498.6	0.880	5.232
15	4.01	4.54	64.9	1.650	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
16	4.44	3.89	64.8	1.645	0.139	.000349	200.0	0.11839	.009721	11.82	11.74	2.62	606.82	100.471	146.81	147718	21752.2	0.158	0.932
17	3.85	3.75	64.8	1.645	0.485	.001163	158.2	0.19693	.030335	36.11	35.89	7.98	1613.97	111.789	47.82	48019	7173.6	0.329	2.122
18	3.44	3.60	64.9	1.650	0.830	.001989	138.2	0.24944	.048868	58.28	57.89	12.88	2602.15	139.491	29.71	29981	4461.9	0.410	2.676
19	2.89	3.71	65.0	1.655	1.389	.003311	113.0	0.31387	.078065	93.17	92.56	20.54	4141.81	161.038	18.48	18486	2826.5	0.580	3.878
20	2.62	3.05	64.8	1.645	1.995	.004771	99.1	0.37890	.110388	150.62	149.22	33.75	6751.48	195.717	14.05	14795	2179.1	0.683	4.581
21	2.36	3.11	64.9	1.650	2.840	.006790	87.6	0.43822	.139886	176.31	174.32	39.23	4381.71	240.836	11.01	11861	1687.8	0.778	5.192
22	2.67	4.58	130.1	2.307	0.063	-	-	-	-	-	-	-	-	-	-	-	-	-	-
23	2.49	4.59	130.0	2.589	0.231	.000397	187.1	0.11839	.009721	11.84	11.78	2.62	364.33	56.193	146.78	148040	21800.9	0.158	0.880
24	2.75	3.42	130.0	2.589	0.672	.001096	158.9	0.19693	.030335	36.13	35.96	7.85	997.41	156.860	48.44	48956	7291.1	0.329	2.179
25	2.35	3.32	129.9	2.582	0.808	.001888	137.8	0.24944	.048868	57.84	57.30	12.75	2907.44	171.694	30.02	30215	4500.0	0.406	2.668
26	2.26	3.14	129.9	2.582	1.398	.003235	117.1	0.31387	.078065	91.18	90.29	19.49	4869.15	199.162	20.68	20796	3219.9	0.569	3.657
27	2.22	2.94	130.0	2.589	2.061	.004757	100.9	0.37890	.110388	148.22	146.82	33.02	6602.69	226.369	14.72	14791	2289.4	0.692	4.372
28	1.91	2.69	130.1	2.597	3.313	.007711	84.7	0.43822	.139886	175.06	172.76	38.63	5771.26	277.526	11.74	11884	1779.3	0.784	4.998
29	2.12	4.06	180.0	3.456	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
30	2.77	3.30	180.0	3.458	0.138	.000320	211.9	0.11839	.009721	11.88	11.82	2.59	658.45	197.894	197.04	197814	27861.9	0.156	0.851
31	2.39	3.32	180.1	3.460	0.466	.001085	160.8	0.19693	.030335	36.32	36.17	8.03	971.68	209.578	47.35	47593	7046.9	0.320	2.081
32	2.17	3.24	180.0	3.458	0.778	.001814	135.7	0.24944	.048868	58.43	58.19	12.93	486.46	221.884	29.56	29977	4486.7	0.400	2.603
33	2.08	3.22	179.8	3.453	1.561	.003621	112.9	0.31387	.078065	93.13	92.74	20.64	4958.39	221.928	18.55	18661	2818.9	0.662	4.308
34	1.86	2.94	180.1	3.460	2.582	.005769	93.5	0.37890	.110388	131.73	131.12	29.16	5781.73	249.592	14.80	14748	1965.9	0.699	4.201
35	1.64	2.72	179.8	3.419	3.731	.008299	77.8	0.43822	.139886	179.99	178.64	39.74	7948.63	291.108	10.87	14829	1980.3	0.717	4.693
36	2.98	3.97	89.7	1.206	1.146	.007666	120.9	0.11839	.009721	11.81	11.75	2.62	70.89	10.722	124.36	17272	16813.0	0.098	0.647
37	1.81	3.75	89.9	1.213	1.647	.009374	107.9	0.19693	.030335	33.89	33.67	7.94	149.72	54.743	40.93	41450	6805.0	0.299	1.781
38	0.98	3.91	89.8	1.208	2.052	.004683	99.0	0.24944	.048868	58.13	57.82	12.85	194.07	82.988	83.85	29376	3815.3	0.597	2.597
39	0.98	7.97	89.8	1.208	2.598	.006636	90.1	0.31387	.078065	93.03	92.57	20.60	326.64	123.914	16.45	16611	2483.8	0.548	3.611
40	0.96	7.84	89.9	1.213	3.237	.007396	84.2	0.37890	.110388	131.55	130.92	29.12	327.72	168.874	11.82	11787	1766.6	0.717	4.704
41	0.95	6.99	89.7	1.204	3.576	.008634	78.0	0.43822	.139886	179.17	178.15	39.68	343.91	221.688	8.93	8695	1299.6	0.903	6.000
42	3.96	3.96	89.6	1.208	0.922	.00222	138.9	0.19693	.030335	36.39	36.18	8.08	485.90	99.899	47.40	47665	7085.1	0.321	2.151
43	4.18	42.8	89.8	1.208	0.482	.00116	154.3	0.19693	.030335	36.26	36.04	8.06	320.69	102.431	47.46	47740	7097.2	0.320	2.171

Fig. 22

FIG. 22.—TABLE II.—(b) RADOJET WITHOUT INTERCONDENSER—HANDLING MOISTURE SATURATED AI

	46	47	
	-	-	
3·2	0.731	0.889	
4·3	0.599	2.101	
7·4	0.617	2.787	
4·9	0.602	4.076	
5·0	0.699	4.681	
4·3	0.886	5.290	
	-	-	
4·9	0.158	0.930	
4·9	0.328	2.146	
4·7	0.661	2.973	
4·8	0.696	4.138	
4·8	0.795	4.656	
4·6	0.800	5.230	
	-	-	
4·8	0.158	0.992	
4·6	0.303	2.102	
4·8	0.660	2.976	
4·5	0.680	3.878	
4·1	0.669	4.398	
4·8	0.778	5.150	
	-	-	
4·9	0.148	0.880	
4·1	0.303	2.179	
4·0	0.695	3.068	
4·2	0.703	3.697	
4·6	0.698	4.173	
4·3	0.726	4.998	
	-	-	
4·5	0.156	0.895	
4·9	0.309	2.189	
4·7	0.680	3.098	
4·9	0.612	4.182	
4·9	0.699	4.986	
4·3	0.717	4.698	
5·0	0.918	0.649	Second Stage only
5·0	0.939	1.789	- - -
4·3	0.987	2.799	- - -
4·8	0.548	3.619	- - -
4·5	0.727	4.766	- - -
4·6	0.909	4.888	- - -
4·4	0.981	2.198	Mean of above readings
4·2	0.939	2.177	Depths in Nature

SATURATED AIR.

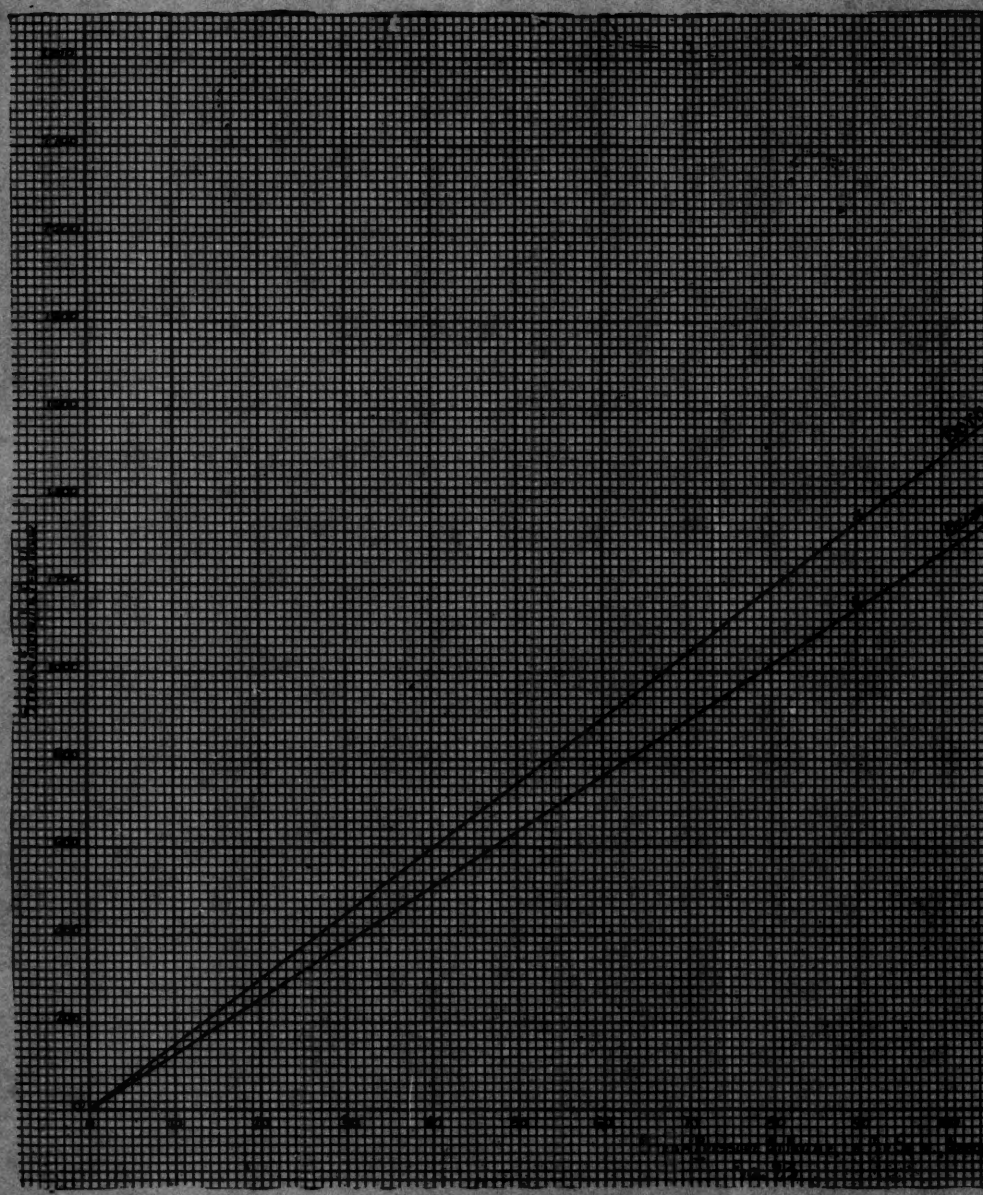
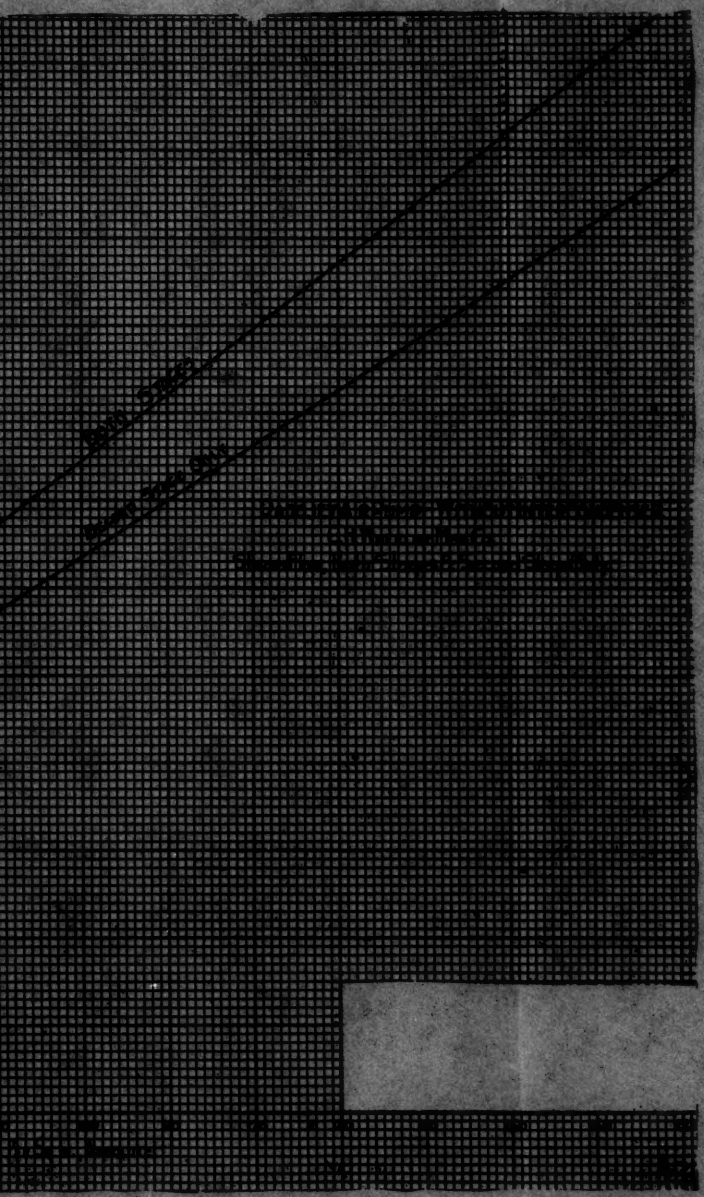


FIG. 23.—STEAM FLOW CURVES—RADOJET WITHOUT INTE



ITHOUT INTERCONDENSER.

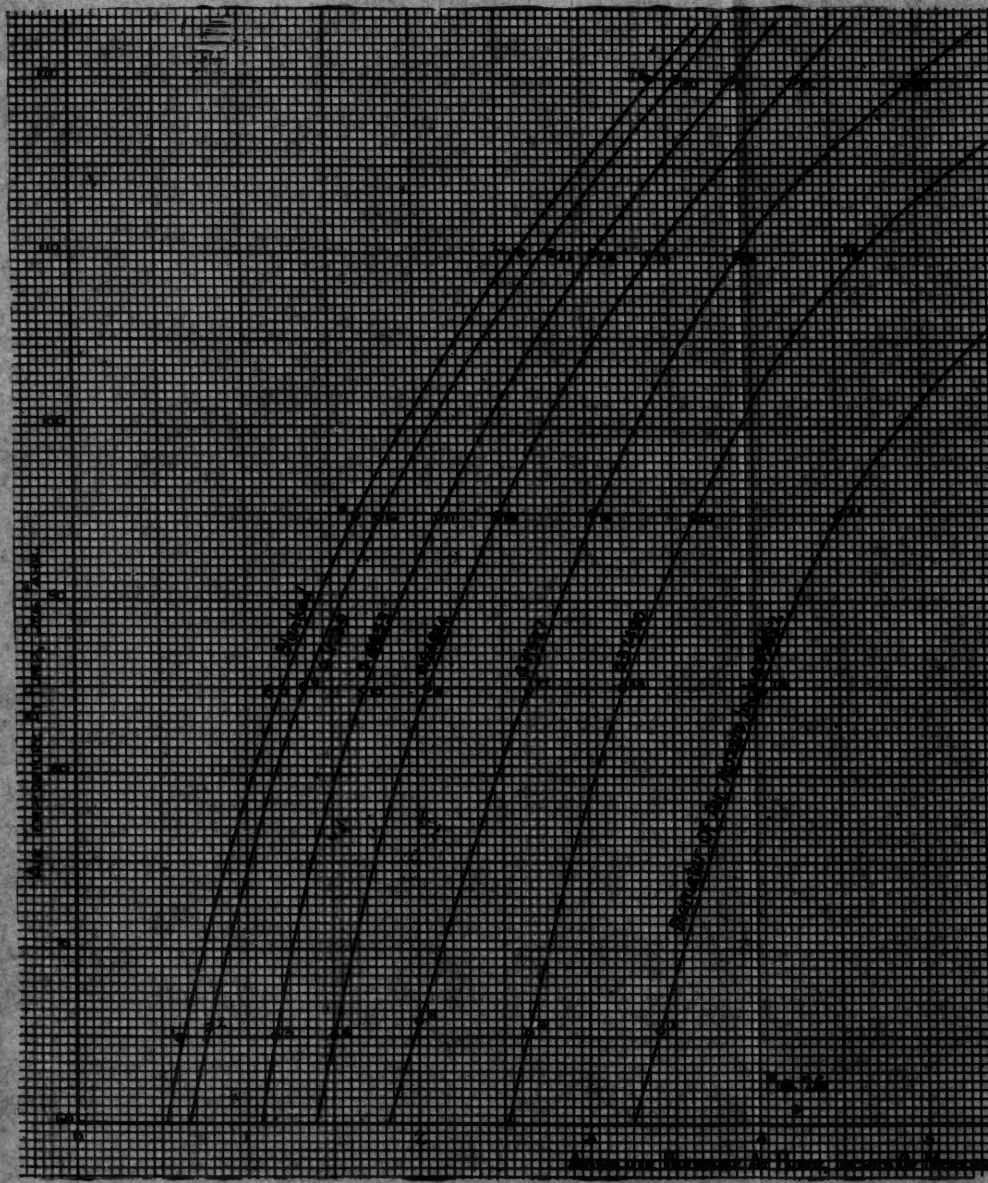
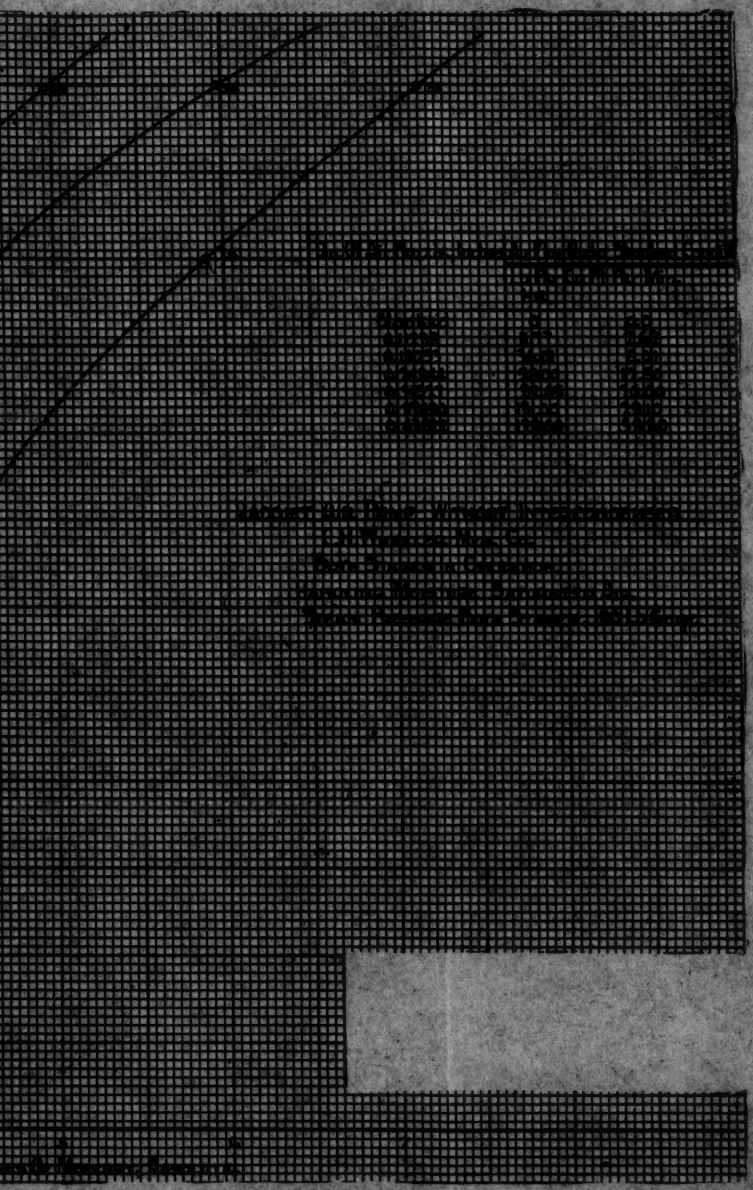


FIG. 26.—CHARACTERISTICS OF RADOJET WITHOUT INTERCONDENSER—H



ONDENSER—HANDLING MOISTURE SATURATED AIR.

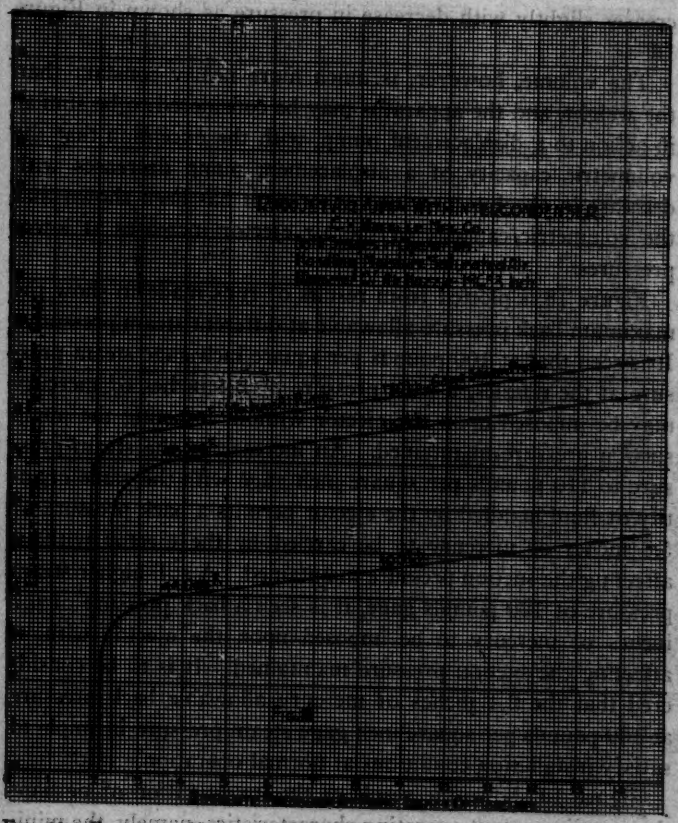


FIG. 18.—EFFECT OF VARYING BACK PRESSURE—INTERCONDENSER RADOJET.

Comparing the volume and weight characteristics for a saturated mixture it is seen that the weight of mixture decreases with a decrease of absolute pressure, almost in direct proportion to the decrease in pressure. The volume of mixture increases slightly with decrease in pressure as shown in Figure 16.

The volume, however, is much more nearly constant than the weight, and the apparatus may be considered to be similar in action to a displacement pump, with the exception that the volumetric capacity of a displacement pump decreases with decrease in absolute pressure, whereas it has been shown that the volumetric capacity of the air ejector increases at lower pressures.

Figure 18 is plotted from results of runs made to show the maximum back pressure allowable at different steam pressures. It is seen that at 100 pounds per square inch gage steam pressure at the nozzles, a back pressure of 2.4 inches of mercury may be imposed upon the ejector without affecting the pump suction pressure maintained. An increase of the steam pressure to 110 pounds per square inch gage with the same quantity of fluid handled by the ejector permits a back pressure of 5.8 inches of mercury to be carried by the ejector without affecting the suction pressure maintained. A still higher steam pressure of 120 pounds gage, it is seen a back pressure of 6.7 inches of mercury to be imposed upon the ejector without affecting the suction pressure maintained. It is to be considered that the 2.4 inches of mercury allowable back pressure at 100 pounds per square inch gage steam pressure is ample to satisfy installation conditions at normal operation.

Figure 19 shows the result of the investigation of perhaps the most important operating characteristics, namely, the minimum allowable steam pressure at the nozzles for safe operation of the ejector. This curve was obtained by increasing the steam pressure until a maximum vacuum was gotten at the pump with the given constant capacity of expanded atmos-

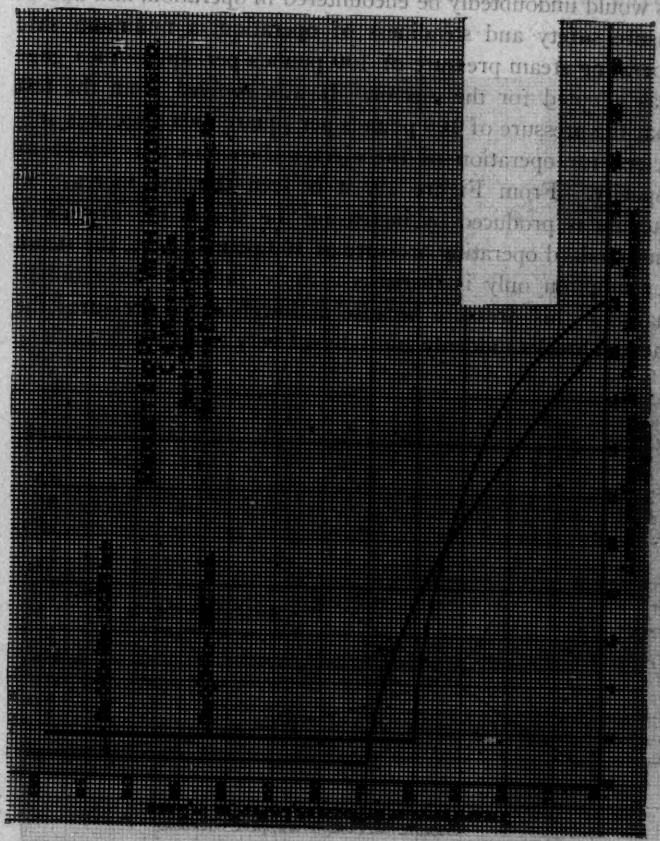


FIG. 18.—CURVE OF CRITICAL STEAM PRESSURE—INTERCONDENSER RADOJET.

pheric air. From the curve it may appear that 95 pounds per square inch gage is sufficient steam pressure at which to operate; but in order to allow for slight back pressure variations as would undoubtedly be encountered in operation, and also to insure safety and steadiness of operation, a minimum safe operating steam pressure of 100 pounds per square inch gage was adopted for the ejector. It was evolved from the test that the pressure of 100 pounds per square inch gage was sufficient for operation of the ejector throughout its range of capacity. From Figure 19, it is seen that no increase in vacuum is produced by increasing the steam pressure above the standard operating pressure of 100 pounds gage, the steam consumption only is increased. It is, however, essential to maintain this pressure because at lower steam pressures the vacuum breaks suddenly and falls off very rapidly.

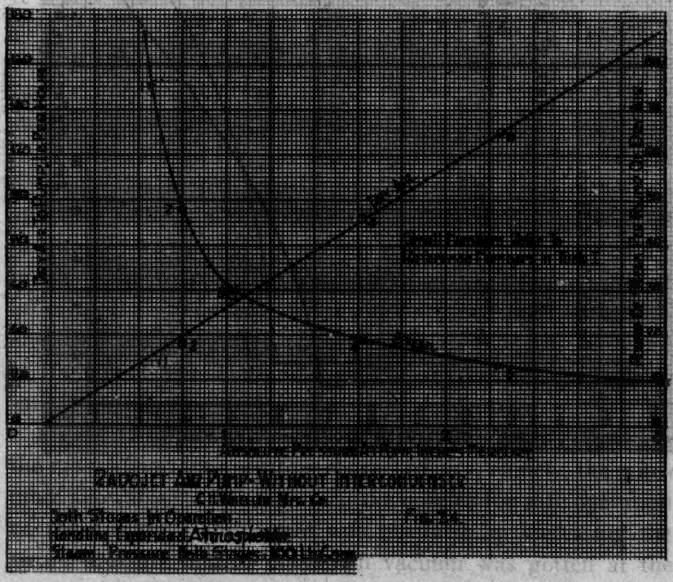


FIG. 24—CHARACTERISTICS OF RADOJET WITHOUT INTERCONDENSER—HANDLING DRY AIR.

The performance of the plain Radojet ejector without inter-condenser, illustrated in Figure 4, is given in Figures 20 to 29 inclusive. The performance of the plain Radojet ejector when compared with that of the inter-condenser Radojet ejector

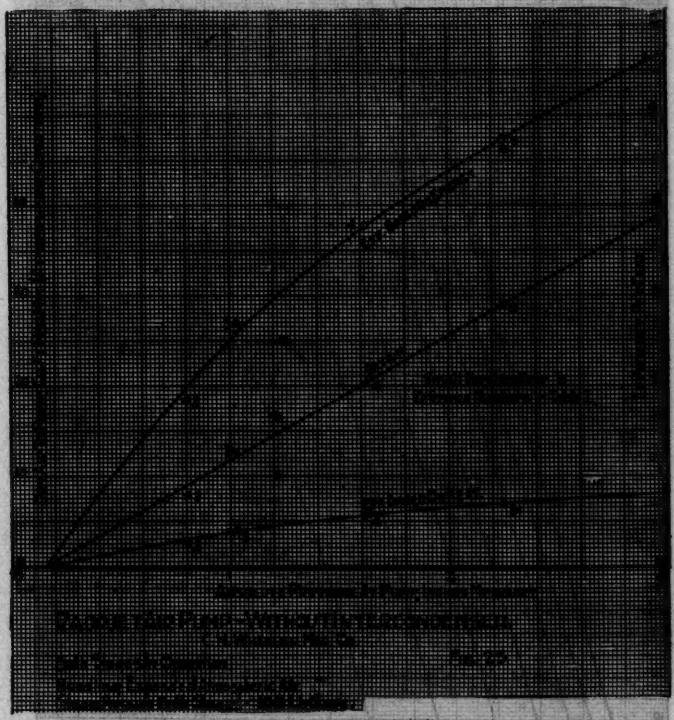


FIG. 25.—CHARACTERISTICS OF RADOJET WITHOUT INTERCONDENSER—HANDLING DRY AIR.

shows that the performances of the two types are very similar; both qualitatively and quantitatively, with the single important exception that the pump without the intercondenser uses 1,720

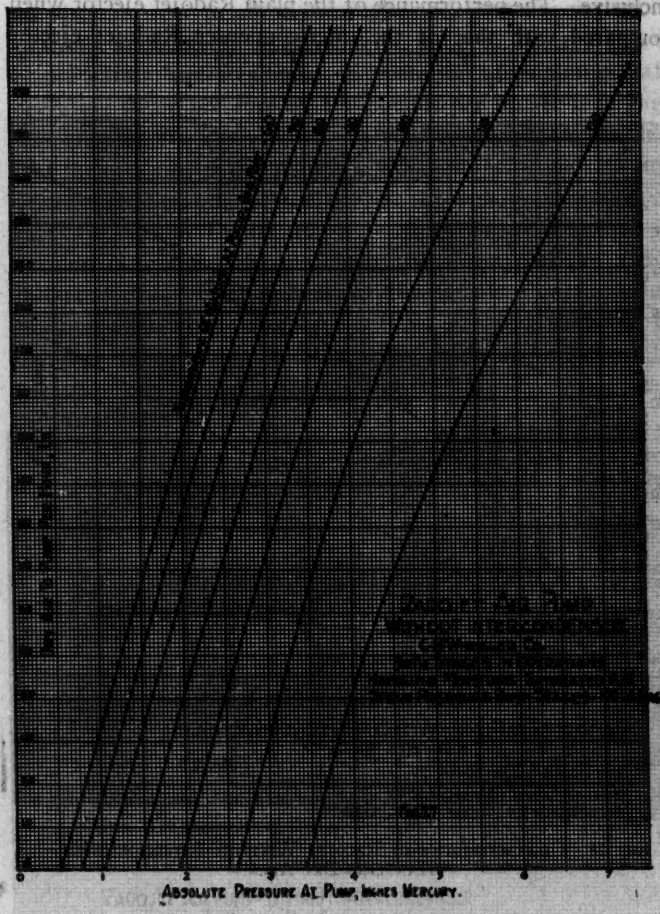


Fig. 27.—Characteristics of Radojet Without Intercondenser—Handling Moisture Saturated Air.

pounds of steam per hour, which is almost exactly twice the 857 pounds of steam per hour used by the ejector with inter-condenser.

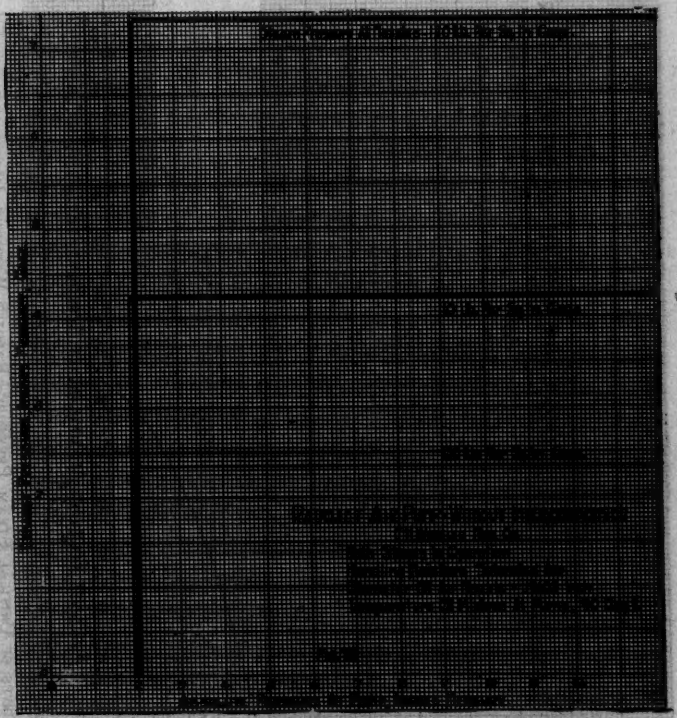


FIG. 28.—EFFECT OF VARYING BACK PRESSURE—RADOJET WITHOUT INTERCONDENSER.

The choice of the type to use in a given installation depends upon whether the exhaust steam is required for heating the feed. By circulating the condensate from the main condenser through the intercondenser shell, the heat in the steam used by the first stage of the inter-condenser pump is recovered, so that the two types of pumps are on the same basis as regards heat

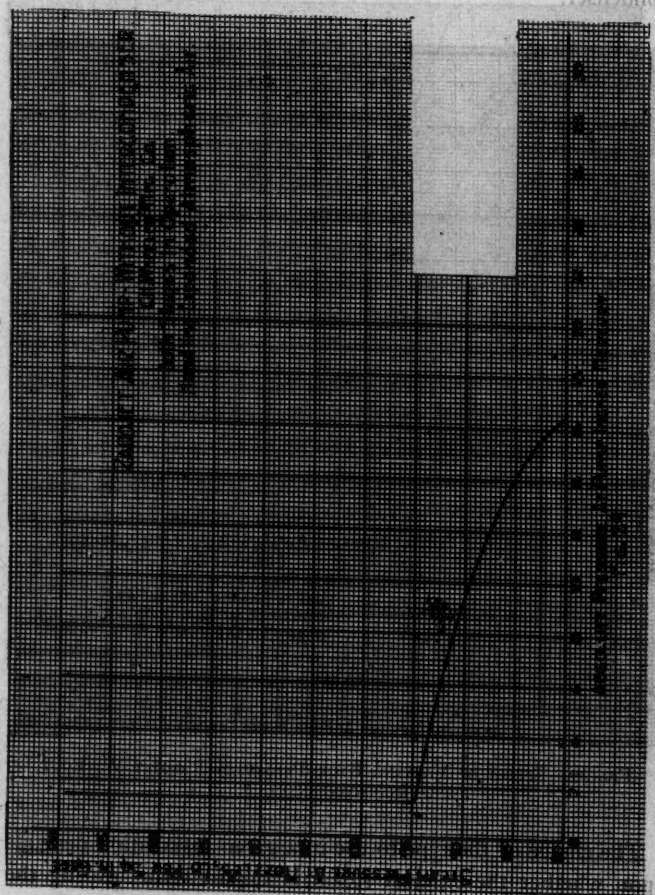


FIG. 39.—CURVES OF CRITICAL STEAM PRESSURE—RADOJET WITHOUT INTERCONDENSER.

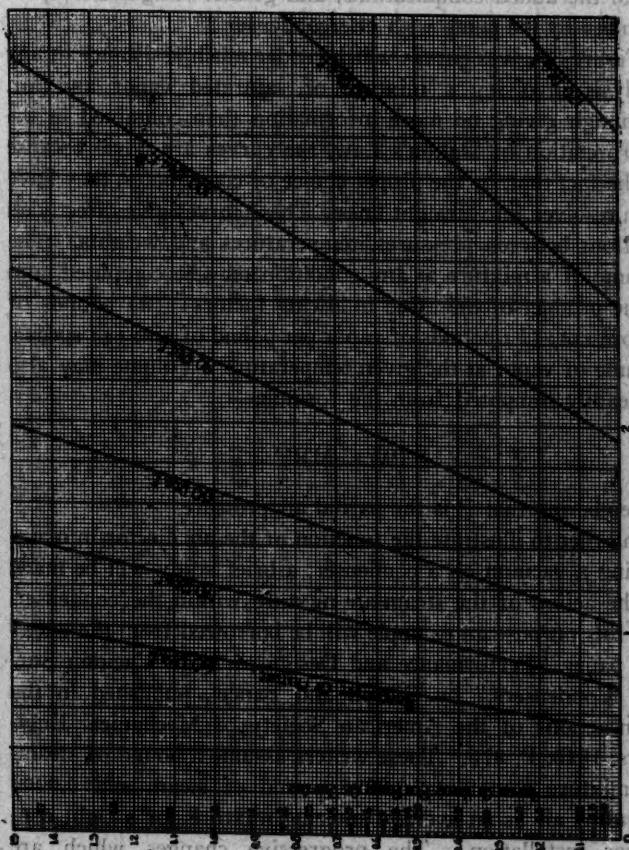


FIG. 30.—RELATIVE WEIGHTS OF AIR AND VAPOR IN A SATURATED MIXTURE.

FIG. 29.—CURVES OF CRITICAL STEAM PRESSURE—RADOJET WITHOUT INTERCONDENSER.

economy. If the exhaust steam may be advantageously used for heating the feed, the pump without intercondenser should be chosen, especially in marine and naval service, in order to avoid the added complications, and greater weight and space required by the pump with intercondenser.

Figure 30 gives curves which render much assistance in any analysis involving a mixture of saturated vapor and air. Although giving no more information than contained in the interpretation of Dalton's Law, the curves are novel because of the fact that, by the proper choice of coordinates, they have been resolved into straight lines. These curves give directly all relations between pressures, temperatures and weights. The volume of a mixture at any condition may be computed, and is the product of the three following factors: (*a*) pounds of air present; (*b*) pounds of vapor per pound of air; (*c*) specific volume of vapor at the given temperature. Volume may also be readily computed from the formula—

$$PV = MRT$$

remembering that P is the partial air pressure.

In order to illustrate the function of the air ejector in maintaining a vacuum on a condenser, a brief study will be made of the action of the steam-air mixture within the condenser. For this purpose an example will be borrowed from the excellent paper, "Performance and Design of High Vacuum Surface Condensers," by Gibson and Bancel, published in the Transactions of the American Society of Mechanical Engineers, 1915, P. 975. These writers assume a pound of steam-air mixture containing 0.9998 pound of steam and 0.0002 pound of air as an average entering mixture for a tight condenser installation. The progressive changes which are brought about in this mixture as it passes through the condenser are tabulated in Figure 31, and explained in the following paragraph, which is quoted with slight modification from the above mentioned paper.

Table showing typical action in a condenser supplied with 1 lb. of steam-air mixture, containing 0.0002 lb. air. Illustrating the process of steam condensation and air compression at a constant vacuum of 29 inches.

1	Temperature of mixture, Deg. F.	79	79	79	76	72	69
2	Partial pressure of steam, in Hg.	1.0	1.0	0.99	0.90	0.80	.50
3	Partial pressure of air, in Hg.	1/5000	1/4000	1/1000	1/100	1/10	1/2
4	Pounds of air present.	.0002	.0002	.0002	.0002	.0002	.0002
5	Pounds of steam condensed.	000	.5000	.8748	.9875	.9987	.99967
6	Pounds of steam present.	.9998	.4998	.1250	.0125	.0011	.00015
7	Volume of air present, cu.ft.	687	328	82	8.2	.825	.164
8	Volume of mixture, cu. ft.	687	328	82	8.2	.825	.148
9	Volume of mixture, cu.ft. per lb.air.	3285000	1640000	410000	41000	4116	820
10	lb. of air per 1000 cu.ft. mixture.	.0003	.0006	.0024	.024	.243	1.22
	NOTE:	Condenser entrance Point A		Steam condensing in this zone.		Air cooling and compressing in this zone.	Air pump suction Point B.

FIG. 31.—TABLE SHOWING CONDENSER ACTION.

At one inch of mercury absolute pressure and a steam temperature of 79 degrees Fahrenheit the volume of 0.9998 pound of steam is 657 cubic feet and the volume of the 0.0002 pound of air is also 657 cubic feet, from which its partial pressure is calculated as 1/8000 inch of mercury. This pressure is so small as to appear almost negligible, and it would seem that such steam could be called air-free. In fact, it is practically air-free steam and acts as such, giving a high heat transmission coefficient so long as the partial pressure of the air remains so small. When half the steam has been condensed, the volume will be half and the partial air pressure 1/4000 inch, causing a corresponding decrease in the partial steam pressure and temperature, as shown by Figure 31. The steam can still be called pure steam. In fact, about 99 per cent of the steam must be condensed before there is any appreciable increase in partial pressure of the air, or appreciable decrease in temperature of the air-steam mixture as shown by Figure 31. When 99 per cent has been condensed, the partial air pressure is then 1/100 inch, but the air is still very rarified and its volume is still too large for removal by a vacuum pump. Thus a process of air compression must take place after nearly all the steam has been condensed, which explains the phenomenon shown in numerous tests in which there is no appreciable rise in circulating water temperature in the bottom of the condenser where the air concentration and compression is taking place. No appreciable amount of steam remains to be condensed, but, still, surface is needed to condense the small amount present, and to cool the mixture, the heat exchange being greatly impeded by the presence of the air. The richer the mixture delivered to the air pump, is in air, the greater the work to be performed by, and hence (other things being equal) the larger this "inactive" zone of the condenser.

Thus it is seen that, in order to remove air from the condenser it is absolutely essential that the temperature at the air pump suction be depressed below the temperature correspond-

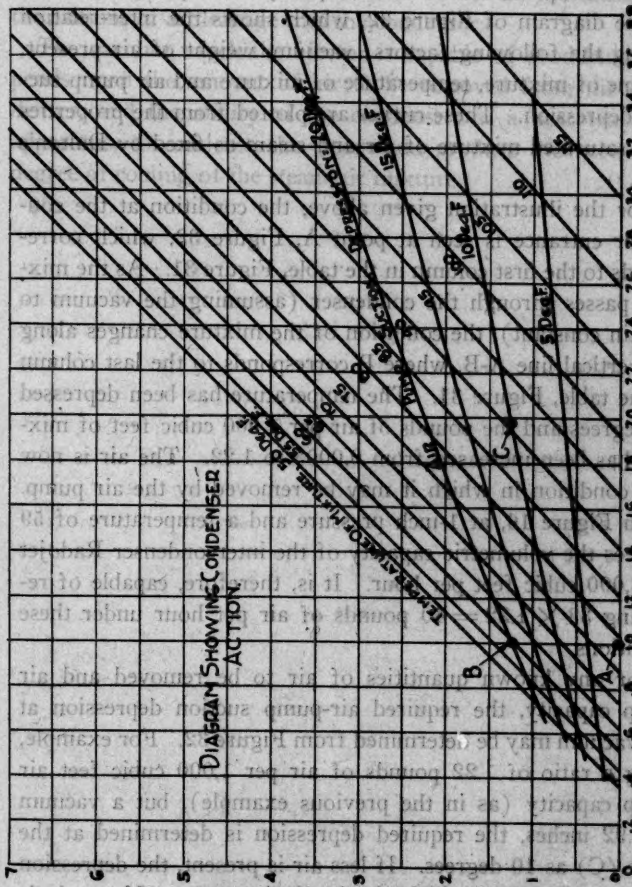


FIG. 32.—DIAGRAM SHOWING CONDENSER ACTION.

ing to the vacuum. Moreover, there is a definite relation between the amount of this depression and the weight of air to be removed per unit volumetric capacity of air pump, as shown by the diagram of Figure 32, which shows the inter-relation among the following factors; vacuum, weight of air present, volume of mixture, temperature of mixture and air pump suction depression. These curves are plotted from the properties of a saturated mixture of air and steam as fixed by Dalton's Law.

For the illustration given above, the condition at the condenser entrance is seen at point A, Figure 32, which corresponds to the first column in the table, Figure 31. As the mixture passes through the condenser (assuming the vacuum to remain constant), the condition of the mixture changes along the vertical line A-B, where B corresponds to the last column of the table, Figure 31. The temperature has been depressed 20 degrees and the pounds of air per 1,000 cubic feet of mixture has been increased from 0.0003 to 1.22. The air is now in a condition in which it may be removed by the air pump. From Figure 16, at 1-inch pressure and a temperature of 59 degrees the volumetric capacity of the intercondenser Radojet is 33,000 cubic feet per hour. It is, therefore, capable of removing $33 \times 1.22 = 40$ pounds of air per hour under these conditions.

For any known quantities of air to be removed and air pump capacity, the required air-pump suction depression at any vacuum may be determined from Figure 32. For example, using a ratio of 1.22 pounds of air per 1,000 cubic feet air pump capacity (as in the previous example), but a vacuum of 1.92 inches, the required depression is determined at the point (C) as 10 degrees. If less air is present, the depression at a given vacuum will be less and vice versa. If no air is present, it will be impossible to produce a depression of the temperature at the air-pump suction.

The depression of the air pump suction temperature, taken alone, is therefore not an indication of efficient action of either the air pump or the condenser, but is rather an indication of the amount of air leakage into the condenser.

The vacuum which the ejector is capable of producing depends upon the amount of air to be removed from the condenser, and the ability of the condenser, as a heat transfer apparatus, to condense the steam and produce the necessary degree of cooling of the steam-air mixture.

FLOW OF STEAM THROUGH PIPES AND FITTINGS.

By M. C. Stuart, Associate, and M. H. Russell.

A considerable amount of energy has been expended by various investigators to discover the laws which govern the flow of steam through pipes. Several years ago Prof. G. F. Gebhardt collected and studied all the existing data on this subject and presented the results of his studies in "Power," June 10, 1907, and later in his "Steam Power Plant Engineering." Prof. Gebhardt found that, although the results of the various investigations were presented in widely different forms, there was marked uniformity in the results of a number of the investigations when the numerous formulæ were compared upon a common basis of notation. From among the various formulæ proposed, one, known as the Carpenter or Babcock formula, appeared to be most satisfactory, and after long usage, has proven to be sufficiently accurate for practical purposes. This formula is

$$P = 0.0001306 \quad \frac{W^2 L \left(1 + \frac{3.6}{d}\right)}{y d^5}$$

in which P is the pressure drop, pounds per square inch; W is the weight of steam flowing, pounds per minute; L is the length of pipe, feet; d is the diameter of pipe, inches; and y is the density of the steam, pounds per cubic foot. While the formula may be used in the above form, it is inconvenient on account of the long computation required. The solution of problems involving the use of the formula is greatly simplified by the aid of charts which have been prepared by H. V. Carpenter and published in "Power," December 17, 1912, and June 10, 1913. Since their publication these charts have had wide usage, and their republication in the present article has been made possible through the courtesy of Mr. Fred R. Low, Editor of "Power."

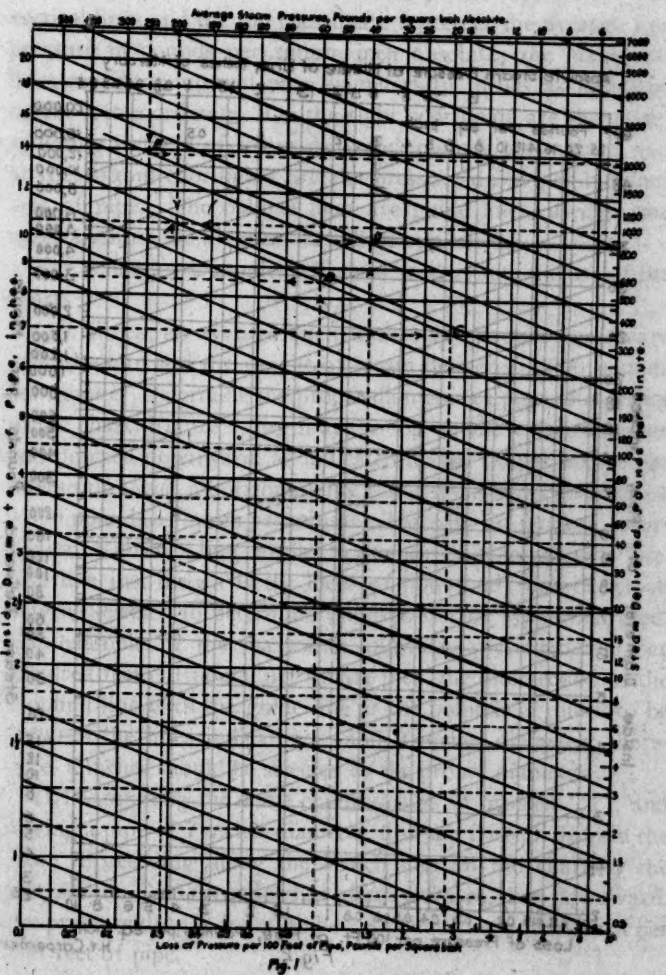

Average Steam Pressures, Pounds per Square Inch Absolute.
Inside Diameter of Pipe, Inches.
Steam Delivered, Pounds per Minute.
Loss of Pressure per 100 Feet of Pipe, Pounds per Square Inch

Fig. 1

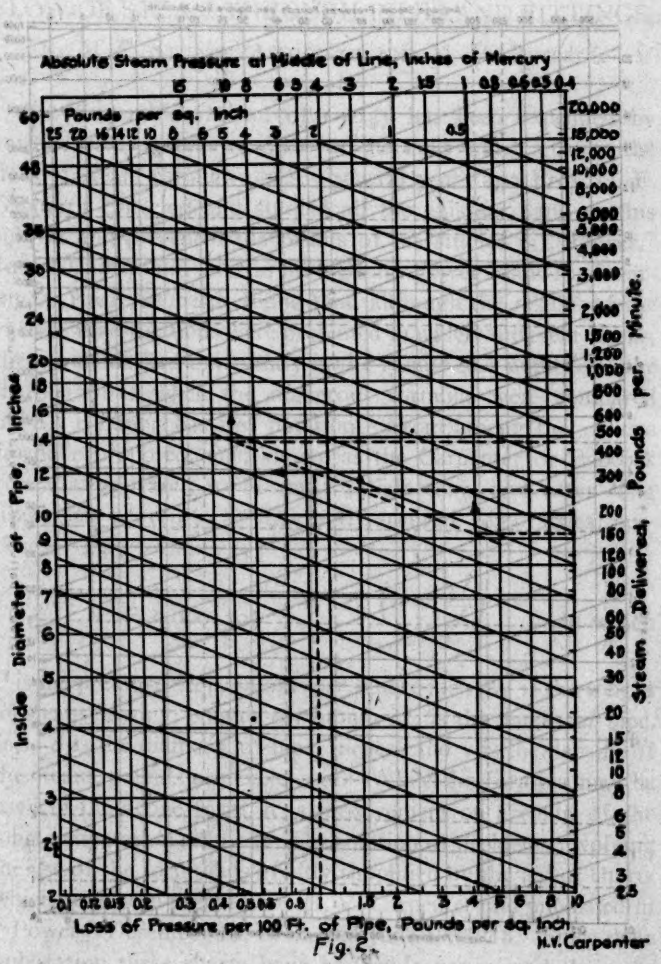

Absolute Steam Pressure at Middle of Line, Inches of Mercury
Pounds per sq. Inch
Inside Diameter of Pipe, Inches
Steam Delivered, Pounds per Minute.
Loss of Pressure per 100 Ft. of Pipe, Pounds per sq. Inch
H.V. Carpenter

Fig. 2.

The Carpenter charts shown in Figures 1 and 2 have logarithmically spaced coordinates on each of the four sides. The vertical lines and the top coordinate represent the average line pressure in pounds per square inch absolute; the horizontal lines represent internal pipe diameters in inches as indicated on the left coordinate. On the right coordinate are the corresponding weights of steam delivered, pounds per minute, and at the bottom coordinate are the pressure drops per hundred feet of pipe. The oblique lines are guide lines, used in the application of the diagram as will be explained.

The following examples will serve to illustrate the use of the charts:*

(1) Given, an average line pressure 200 pounds per square inch absolute and the weight of steam delivered 1,000 pounds per minute. Determine the proper diameter of pipe. Referring to the chart, Figure 1, begin at the top on the 200-pound line and proceed downward, as indicated by the dotted line to the intersection with a horizontal line (shown dotted) representing 1,000 pounds of steam per minute. Through the point of intersection A, draw a line parallel to the nearest oblique guide line. The line just drawn is the locus of points any one of which may be selected to determine the diameter of pipe to be used. For example, if the maximum allowable pressure drop per hundred feet is 1 pound per square inch, the intersection of the 1-pound line with the guide line at the point B is found to be between the lines indicating 8 and 9 inches diameter; therefore, 9 inches should be selected as the proper diameter.

(2) Assuming the same conditions as in problem (1), and having a pipe of 7 inches diameter, find the friction drop in the line. Proceeding along the 7-inch line on the chart to the intersection C on the guide line already drawn, then downward, the pressure drop is found to be 2.8 pounds per square inch per 100 feet of pipe.

* In case it is desirous to use the charts for superheated steam the pressure should be chosen such that the density of the corresponding saturated steam is the same as that of the superheated steam. This is the best procedure available in the absence of experimental data on the friction loss with superheated steam.

(3) A 10-inch line carries steam at 250 pounds per square inch absolute; assuming an allowable loss of 1.5 pounds pressure drop per hundred feet of pipe, find the quantity of steam delivered. Beginning at the bottom of 1.5-pound, project vertically up to the 10-inch line and draw the guide line through the intersection D. The 250-pound line intersects the guide line at the point E directly opposite from which the weight of steam is determined at slightly over 2,000 pounds per minute.

Figure 2 is a chart similar to Figure 1, but for use in the design of low pressure lines. Frequently pipe lines are designed by using certain standard steam velocities as a basis. Although this is a convenient time-saving method, it can be shown to be illogical. Because of the increase in density, the velocity head increases with the pressure for a constant velocity. And as the pressure drop is proportional to the velocity head, it also increases with the pressure at constant velocity. The logical method, then, is to design for a desired pressure drop for the quantity of steam and the pressure existing.

The subject of pipe fittings, bends, sudden contractions and expansions, and valves requires special attention. The simplest method of dealing with these conditions is to solve for the losses in terms of the velocity head. This end is accomplished by making use of the hydraulic formula $h = \frac{V^2}{2g}$ where h is the head in feet of steam, V is the velocity in feet per second and g is the acceleration of gravity. However, the formula in this form cannot be used to advantage in dealing with vapors, therefore, the velocity head is converted into equivalent pressure head in pounds per square inch. The steps involved in making this conversion are shown in detail in Figure 3, in which formula (6) gives velocity head, expressed as pounds per square inch, in terms of weight of steam per unit area and density. Figure 4 is the graphical representation of formula (6) for various constant pressures. It was also found con-

FORMULAE CONNECTING VELOCITY, VELOCITY-HEAD, FLOW, DENSITY, PIPE-SIZE, ETC.

W = Steam flow, lb. per min.
Q = " " cu. ft. per sec.
d = Internal diameter of pipe, in.
a = " area of pipe, sq. in.
A = " " " " sq. ft.
v = Velocity of steam, ft. per sec.
V = " " " ft. per min.
γ = Density of steam lb. per cu. ft.
h = Head, equivalent to velocity, ft. of steam.
H = " " " lb. per sq. in.
P = Pressure drop, lb. per sq. in.
L = Equivalent length of pipe ft.
l = " " " in.

Determination of velocity head in terms of steam flow, density and pipe area.

(1) $h = \frac{v^2}{2g}$

(2) $Q = Av$; $v = \frac{Q}{A}$; $A = \frac{0.7854 d^2}{144}$

(3) $W = 60 \gamma Q$; $Q = \frac{W}{60\gamma}$; substituting value of A and Q

$v = \frac{W \times 144}{60 \gamma \times 0.7854 d^2} = \frac{3.6857\ W}{\gamma d^2}$; $v^2 = \frac{2.3273\ W^2}{\gamma^2 d^4}$

(4) $H = \frac{\gamma h}{144}$ $h = \frac{144 H}{\gamma}$; substituting in (1), $\frac{144 H}{\gamma} = \frac{2.3273 W^2}{\gamma^2 d^4}$

(5) $H = \frac{2.3273\ W^2 \gamma^2}{144 \gamma d^4 \times 2g} = \frac{2.3273 W^2}{144 \gamma d^4 \times 2g} = 0.001008 \frac{W^2 \gamma}{d^4}$.

$\gamma = \frac{W}{A v}$; $V^2 = \left(\frac{W}{A}\right)^2 \frac{1}{\gamma^2}$; $v^2 = \left(\frac{W}{A}\right)^2 \times \frac{1}{\gamma^2} \times \frac{1}{3600}$

$\frac{144 H}{\gamma} = \frac{v^2}{2g} = \left(\frac{W}{A}\right)^2 \frac{1}{\gamma^2} \times \frac{1}{2g} = \frac{1}{3600}$; $H = \left(\frac{W}{A}\right)^2 \times \frac{1}{\gamma} \times \frac{1}{2g} \times \frac{1}{3600 \times 144}$

(6) $H = \left(\frac{W}{A}\right)^2 \times \frac{1}{\gamma} \times \frac{1}{33,338,000}$

Determination of the length of pipe to produce friction head equivalent to the velocity head

(7) $P = 0.0001306 \frac{W^2 L \left(1 + \frac{3.6}{d}\right)}{\gamma d^5}$ = frictional pressure drop. (R.C. Carpenter)

Equating (5) and (7). $0.001008 \frac{W^2 \gamma}{d^4} = 0.0001306 \frac{W^2 L \left(1 + \frac{3.6}{d}\right)}{\gamma d^5}$

(8) $L = \frac{0.001008}{0.0001306} \times d \times \left(1 + \frac{3.6}{d}\right) = 7.718 \left(d + 3.6\right)$

$l = 93 \frac{d}{1 + \frac{3.6}{d}}$ (9) $\frac{l}{d} = \frac{93}{1 + \frac{3.6}{d}}$

Frictional losses due to 90 deg. bends and globe valves in equivalent lengths of pipe.

(10) $l = \frac{76\ d}{1 + \frac{3.6}{d}}$ for bends.

(11) $l = \frac{114\ d}{1 + \frac{3.6}{d}}$ for globe valves. } Briggs.

(12) $l = \frac{93\ d}{1 + \frac{3.6}{d}}$ assuming entire loss of velocity head.

Fig. 3.

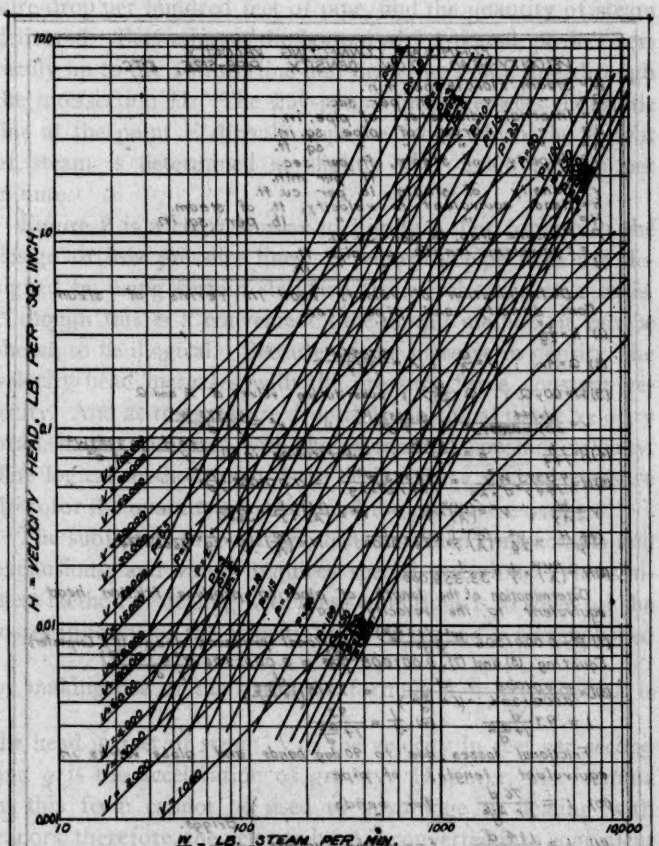


Fig. 4 CURVES OF VELOCITY AND VELOCITY-HEAD IN STEAM LINES AT VARIOUS PRESSURES.

venient to plot on Figure 4 curves of constant velocity which have the same coordinates as the curves of constant pressure.

Losses in bends and valves can only be approximated owing to the large variety of designs, and formulæ for small sizes of

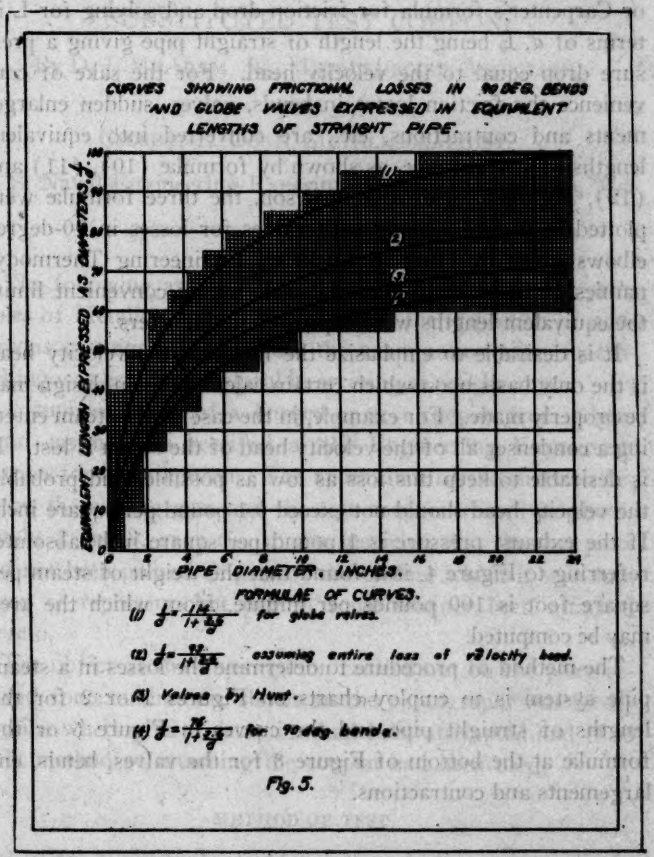

CURVES SHOWING FRICTIONAL LOSSES IN 90 DEG. BENDS AND GLOBE VALVES EXPRESSED IN EQUIVALENT LENGTHS OF STRAIGHT PIPE.
EQUIVALENT LENGTH EXPRESSED AS DIAMETERS, $\frac{l}{d}$
PIPE DIAMETER, INCHES.
FORMULAE OF CURVES.
(1) $\frac{l}{d}=\frac{150}{1+\frac{3.5}{d}}$ for globe valves.
(2) $\frac{l}{d}=\frac{90}{1+\frac{3.5}{d}}$ assuming entire loss of velocity head.
(3) Values by Hunt.
(4) $\frac{l}{d}=\frac{75}{1+\frac{2.5}{d}}$ for 90 deg. bends.

Fig. 5.

fittings are not applicable to large sizes. General formulæ as proposed by Briggs in Gebhardt's "Steam Power Plant Engineering" are given on Figure 3. Losses due to sudden en-

largements and contractions are taken care of by assuming a loss of the entire velocity head in each case. The length of pipe in which the friction loss is equivalent to the velocity head is obtained by equating equation (5) Figure 3, to equation (7) or Carpenter's formula for friction drop and solving for L in terms of d, L being the length of straight pipe giving a pressure drop equal to the velocity head. For the sake of convenience the friction losses in bends, valves, sudden enlargements and contractions, etc., are converted into equivalent lengths of straight pipe, as shown by formulæ (10), (11) and (12), Figure 3. For a comparison, the three formulæ were plotted in Figure 5 along with values for losses in 90-degree elbows given by Hunt in Lucke's "Engineering Thermodynamics." In order to keep the scale within convenient limits the equivalent lengths were expressed in diameters.

It is desirable to emphasize the fact that the velocity head is the only basis upon which certain calculations in design may be properly made. For example, in the case of the steam entering a condenser all of the velocity head of the steam is lost. It is desirable to keep this loss as low as possible, and probably the velocity head should not exceed 0.1 pound per square inch. If the exhaust pressure is 1 pound per square inch, absolute, referring to Figure 4, it is found that the weight of steam per square foot is 100 pounds per minute, from which the area may be computed.

The method of procedure to determine the losses in a steam pipe system is to employ charts on Figures 1 or 2 for the lengths of straight pipe and the curves in Figure 5 or the formulæ at the bottom of Figure 3 for the valves, bends, enlargements and contractions.

METALLOGRAPHY IN THE DETECTION OF A SUBSTITUTED TEST COUPON.

By D. J. McAdam, Jr., Metallurgist, Associate,

AND

Gilbert E. Doan, Metallographist.

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The following article illustrates the application of the principles of Metallography to a problem in the inspection of nonferrous castings. In this case the examination after macroetching was especially helpful in arriving at conclusions.

A supposed case of a burned-on test coupon from a gun metal casting was received at the Engineering Experiment Station for investigation. The appearance of the junction between the casting gate and the test coupon, shown in photographs 1 and 2, gave rise to the suspicion that the test piece had not been poured as part of the casting which it was supposed to represent, but had in some manner been attached thereto.

The object of the investigation carried out on the piece received for examination, shown in the photograph, was to determine whether such deception had been attempted and if so what method of attaching the coupon had been used.

METHOD OF TEST.

Samples from the horizontal and vertical parts of the gate and from the attached test piece were analyzed chemically. The gate was cut into smaller pieces for metallographic examination. The metallographic specimens were etched and

arranged in order so as to show the grain structures in a longitudinal section through the entire gate. This section was photographed unmagnified. Some specimens were also photographed at various magnifications so as to show the typical structures in various parts of the gate and test piece.

RESULTS OF CHEMICAL ANALYSIS.

The results of chemical analysis of drillings from the test piece and from the gate are given in Table 1. The test piece contains more zinc and a much smaller percentage of lead than the gate.

TABLE 1.

Material.	Test Piece Specimen.	Gate of casting.	
		Longitudinal portion.	Vertical portion.
		Specimen 20	Specimen 11
Copper	88.71	88.93	88.15
Zinc	1.90	1.02	1.30
Tin	9.66	9.46	9.70
Lead	0.03	0.25	0.30
Iron	0.01	Trace	Trace.

RESULTS OF SURFACE INSPECTION.

As shown in photographs, 1 and 2, the test piece which includes specimens 1 to 8 and parts of specimens 9 and 10, is enveloped at its junction by metal of the upright part of the gate. This appearance of the junction of the test piece and gate is the reason for the suspicion that the test piece may have been burned onto the casting.

RESULTS OF MACRO-ETCHING.

On account of the large grain size in such material, the shape and orientation of the grains are revealed better by study of the "macrostructure" than by study of the microstructure. The macrostructure of a metal is the structure brought out by "macro-etching," which consists in the etching of a comparatively large surface so that the crystalline structure and possibly segregation may be observed on the unmagnified surface or at low magnifications.

The results of such an examination of the material in question are illustrated by the photographs, 3 and 4. The specimens there shown had been polished and etched with chromic acid. The specimens illustrated in Figure 1 of photograph No. 3 are arranged in the same order as in the uncut piece shown in photograph No. 1. Figure 1 represents essentially a longitudinal section through test piece and gate. Figure 2 shows three of the same specimens that are shown in Figure 1, and also a transverse section through the vertical part of the gate.

Figure 1 of photograph 3 shows the effect of rate of cooling on the grain size of the metal. In the thickest part of the metallic mass, as illustrated by specimens 13A, 14 and parts of specimens 17 and 18, the grain size is very large; this is due to the relatively slow cooling of this part of the casting. The average grain size diminishes with departure in every direction from the center of the upright part of the gate.

At the junction of the test piece and gate, however, as shown on specimen 9 of Figs. 1 and 2, photograph 3, there is an abrupt change in the orientation of the grains. This is illustrated more clearly by photograph 4, which shows specimen 9 at a magnification of about 4½ diameters. It will be noted that the grains of the casting gate are elongated in a direction perpendicular to the junction of the gate and test piece. The grains of the test piece near the junction, however, as observed in this and other sections, are elongated

radially from the axis of the test piece as center. In general, there is abrupt change of structure and in some regions actual discontinuity at the junction of the test piece with the gate. The significance of the structures revealed by macroscopic etching will be discussed below.

RESULTS OF MICROSCOPIC EXAMINATION.

Typical microstructures of test piece, casting gate, and the junction of the two are illustrated by photographs Nos. 5 to 13.

Photographs 5 and 6 show typical microstructures of the test piece at a distance from the junction of test piece and casting gate. The structure is typical of Composition "G" as cast. The dark, dendritic cores outline the crystallites which were the first to solidify after the metal was poured. The orientation of these dendritic cores is uniform throughout each grain. Portions of several grains can thus be distinguished in each of these photomicrographs. Between the crystallites of the alpha solid solution, a few particles of the alpha-delta eutectoid are visible at higher magnifications.

Photographs, 7, 8 and 9 show typical structures found in the casting gate at a distance from the junction with the test piece. The micro-structure is similar to that of the test piece. The size of the grains and of the crystallite cores, varies with the location in the casting gate. In regions near the thicker parts of the gate, as illustrated by photograph 7, the grain size is much larger than in thinner parts, or parts near the surface as illustrated by Figures 8 and 9.

Photographs 10, 11, 12 and 13 show regions along the line of junction of casting gate and test piece on the section shown in photograph 4. An abrupt change of microstructure and in some regions actual discontinuity of the metal are found along this junction. It will be noted that the metal of the test piece preserves the dendritic, cored structure so noticeable in other parts of the test piece as illustrated by photographs 1 and 2. The metal of the casting gate, however, re-

PHOTO 1

PHOTO 2

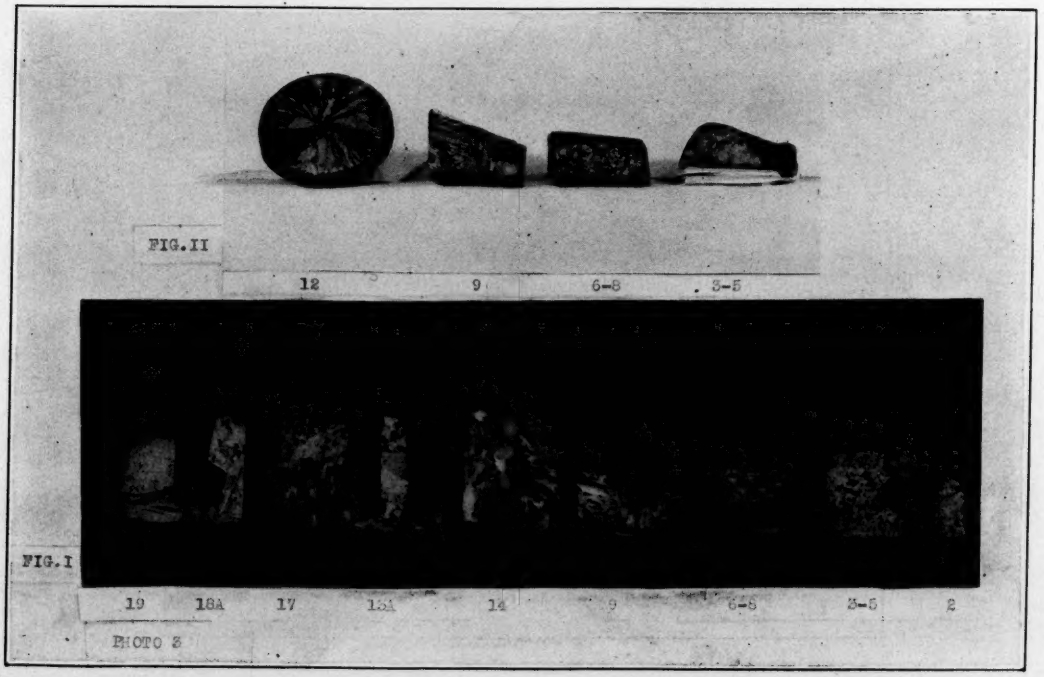

FIG. II

12 9 6-8 3-5

FIG. I

19 18A 17 16A 14 9 6-8 3-5 2

PHOTO 3

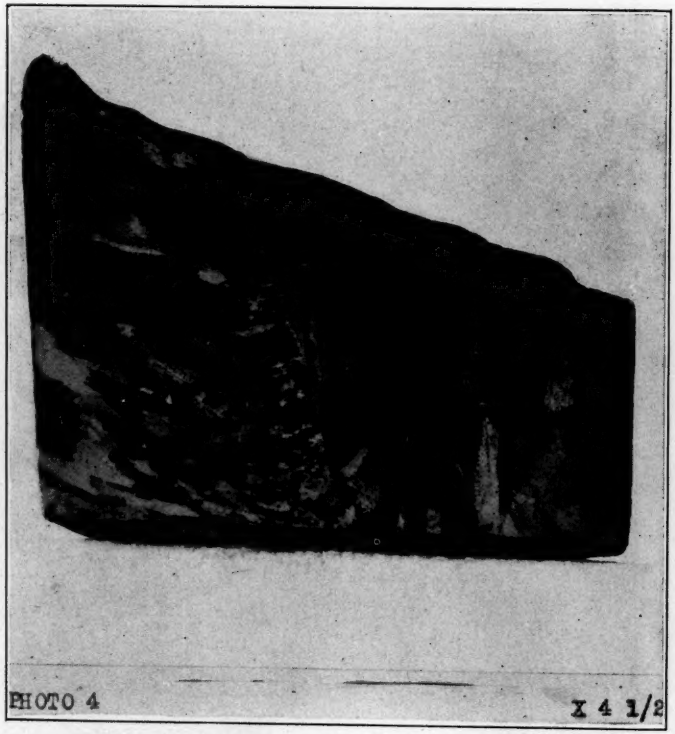

PHOTO 4 X 4 1/2

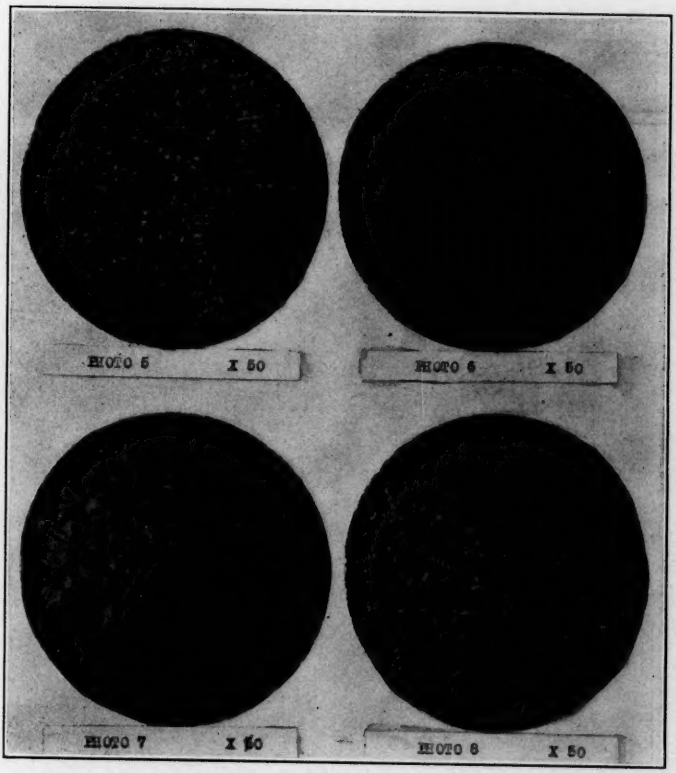
PHOTO 5 X 50
PHOTO 6 X 50
PHOTO 7 X 50
PHOTO 8 X 50

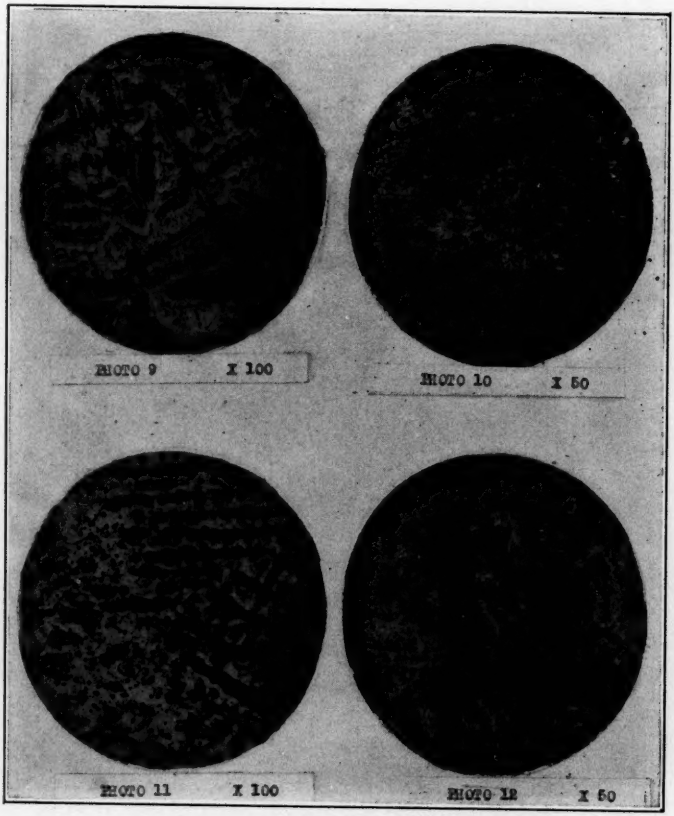
PHOTO 9 X 100
PHOTO 10 X 50
PHOTO 11 X 100
PHOTO 12 X 50

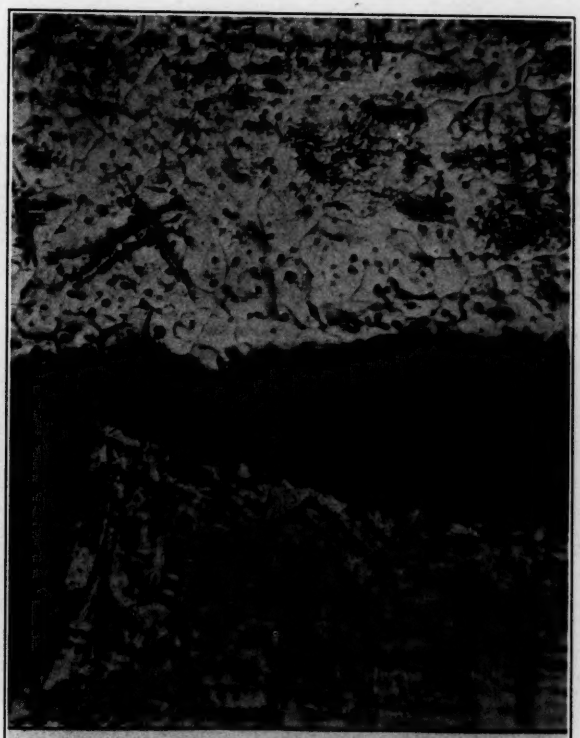

PHOTO 13 x 100

tains at the junction little or none of the dendritic structure; the grain size moreover is much smaller than in the test piece. The microscopic grains at the junction, as well as the macroscopic grains extending away from the junction are elongated in a direction perpendicular to the surface of junction. The significance of this microstructure, with that of the macrostructure will be discussed below.

DISCUSSION OF CHEMICAL COMPOSITION.

Chemical analysis has shown that the casting gate and test piece differ in regard to their zinc and lead content. These differences are sufficient to indicate that the gate and test piece have not been poured from the same melt of metal. The test piece is somewhat superior to the gate in composition.

DISCUSSION OF RESULTS OF MACRO-ETCHING AND METALLOGRAPHIC EXAMINATION.

As shown in the photographs of specimen 9, photographs 3 and 4, there is an abrupt change in the orientation of the grains at the junction of gate and test piece. Photographs 10 and 11 show that there is also an abrupt change in microstructure and in some regions discontinuity of the metal, at the junction of gate and test piece. The results of macro-etching and metallographic examination as well as the results of chemical analysis therefore indicate that the test piece has not been poured as part of the casting it was supposed to represent.

DISCUSSION OF METHOD USED IN JOINING TEST PIECE AND CASTING GATE.

Since the casting and test piece have evidently not been poured at the same time and from the same melt, the question arises in regard to the method used in joining the test piece to the casting gate. Was the test piece burned on to the casting or was some other method used? In regard to this ques-

tion the results of macro-etching and metallographic examination lead to definite conclusions, which are based on the laws of crystallization and grain growth.

The crystalline grains that form during the solidification of a liquid tend to grow in a direction normal to the nearest cooling surface. In other words, the grains grow from the cooler toward the hotter parts of the liquid and are elongated in the direction of the temperature gradient. For example, in the nearly cylindrical upright part of the casting gate, as illustrated by specimen 12 of photograph 3, the grains have grown from the circumference toward the axis, thus producing radial elongation. In parts of the casting, in which the temperature gradient during solidification was slight, the grains are nearly equi-axed. This is illustrated by specimens 3-5 of photograph 3.

As illustrated by specimen 9 of photographs 3 and 4 the direction of elongation of the grains in the casting gate is nearly perpendicular to the surface of junction of gate and test piece. Evidently the grains have grown from this surface of junction toward the center of the upright part of the casting gate. The surface of junction could not have served thus as a chilling surface unless the solid test piece had already been in place when the casting gate was poured. The test piece, therefore, was joined to the casting by pouring the casting with the solid test piece already in position in the mold.

Contact with the solid test piece evidently chilled the molten metal of the casting gate below the point of solidification, while the test piece itself was not melted even superficially. The grain size, however, of the test piece near the junction has been enlarged considerably by the heat from the molten casting gate. This will be evident from a comparison of the grain sizes of specimen 9 and specimen 6-8 of the test piece.

The results of metallographic examination, as illustrated by photographs 10, 11, 12 and 13, confirm the evidence obtained

by macro-etching. The microscopic grains of the casting gate near the junction with the test piece have grown in a direction normal to that junction. The absence of cored structure in these grains is due to the chilling contact with the solid test piece. The grains have solidified so quickly that sufficient time was not available for selective freezing.

CONCLUSIONS.

Chemical analysis has shown that this casting gate and test piece are not from the same melt of metal.

Macro-etching and metallographic examination have shown that the casting and test piece were not poured at the same time. In other words, the test piece was not poured as part of the casting that it was supposed to represent.

Macro-etching and metallographic examination have shown that the solid test piece was already in position in the mold at the time when the casting was poured. The junction between test piece and casting gate, therefore was obtained by placing a solid test piece in the mold and afterward pouring the casting.

BETTER BOILER WATER.

By Commander C. A. Bonvillian, U. S. N., Member.

In the operation of the Engineering plant of a ship, there is nothing more important, or that requires more constant attention than boiler water. The maintaining of good quality water in the boilers, and the prevention of impurities from entering feed water, is a problem that is continually facing the engineering personnel. The more care and intelligence given to this matter, the better will be the results obtained with the boiler plant, which will be favorably reflected in the general efficiency of the entire plant. While considerable advancement has been made in the proper care of water for boilers, during the past few years, and the present practices are considered generally sound, there is still a good deal more that can be done to improve the general operating conditions by a stricter and more intelligent enforcement of the present practices, and also, by further improvement and development of these practices. The principal advancement in the care of water for boilers resulted from the development several years ago of the accurate chemical test for salinity and alkalinity at present in use.

The subject of boiler water has increased in importance and considerable improvement has resulted in practices on board ship with the requirements demanded by the trend in boiler design and in the use of oil as fuel. Oil burning boilers are so frequently and readily forced to a high degree, that unless a good quality water is maintained, quick deterioration or damage will result. With the increasing of the rate of combustion per square foot of heating surface in boiler design, it is evident that the maintaining of a much higher quality water than is at present demanded becomes an important

necessity. The use of smaller tubes in boilers also demands a better quality of boiler water. We then see that the trend of design is leading to a demand for purer boiler water in order to get satisfactory results.

One of the difficult things about improvement in the practice of boiler operation is that there is no other subject on board ship into which the human element enters so largely. As a result of this, there are bound to be wide divergencies in the results obtained. The human element being so important in this respect, the real road to improvement is in the education of the personnel along proper lines, necessary restrictions to make all ships conform strictly to the best practices, frequent efficient inspections to see that best practices are being followed, and analysis of the operation records at intervals to determine the results obtained in comparison with other ships. Unless water tests are intelligently and carefully made as required, and unless these results are intelligently applied, full advantage is not being obtained of the present knowledge of this subject. An analysis of the history of boilers in service of the same age and service will show a large divergence in the life, condition of the water sides, and cost of repairs, which can be directly traceable to the variable degree of care given by the personnel to boiler water.

Another difficulty that is an obstacle to the improvement in practices is that carelessness does not generally make itself felt immediately, but at some distant date in the future when the personnel really responsible may have long since left the ship, this of course depending on the degree of carelessness allowed to exist.

It is well known that the important considerations in boiler operation are: 1—Prevention of Deterioration. 2—Economy. 3—Safety. 4—Reliability.

It can be readily seen that the quality of the boiler water affects all these factors in operation. As the life of a boiler depends upon the rate of progress of corrosion, the preven-

tion or delay of this destructive action is one of the most important duties of the engineer personnel on board ship. It is well known that the use of pure boiler water will result in a minimum of corrosion. Its use will therefore increase the life of the boiler, reduce upkeep cost materially, give better economy, give the desired reliability, and maintain that safety to the pressure parts which is so essential. It is therefore an essential and highly important matter to strive for pure boiler water. Pure boiler water means water free from impurities and since it is impossible to arrive at perfection, this matter devolves itself into one of degree of purity.

The principal impurity encountered in boiler water on board ship is sea water, which of course is the ever present bugbear of the engineering personnel. Sodium Chloride which is the principal component of sea water has a marked effect on corrosion. This effect varies with the Sodium Chloride content, or salinity in the boiler water, and, unless the water is kept properly alkaline, will in short order produce serious damage to the water sides of a boiler. As this salinity increases it becomes more difficult to maintain the alkalinity at the proper amount, so with high salinity there is the additional hazard of damage from this cause. The reduction of salinity will give a proportionate reduction in corrosion, so that the best results are obtained with boilers that are maintained with a low salinity.

Another source of impurity in boiler water is the fresh waters obtained on shore in the various ports of the world. Water from shore contains impurities of varying composition and degree depending on the nature of the soil in the territory through which it has passed, these impurities being the different carbonates, sulphates, chlorides, vegetable acids, mud, air and gases.

The source of water for boilers on board ship is by distilling, and by obtaining water from shore. The principal source is from distilling by means of the evaporating and

distilling plant. If high quality water is made by efficient operation this water is generally of higher quality than can be obtained from shore. There is considerable variation in what is thought to be high quality distilled water and insufficient attention has been paid to this point. It is considered that water can be distilled very readily on Naval Ships of 0.7 grain of chlorine per gallon or less and water of this analysis or less is considered very high quality water and more suitable for boilers than shore water. Results have demonstrated that, with good operation, water carefully distilled is generally about 0.3 grain of chlorine per gallon or less and hardly ever going over 0.5 grain of chlorine per gallon. How many ships pay attention to this point and try to get this quality water for make up feed? Generally no attention is paid to this point and they do not know the exact quality of water they are making, except when the matter comes to attention due to its poor quality.

While the analysis of the shore waters may show a good many to be sufficiently pure for use in boilers, the great majority of them available at the different ports are not suitable and should be avoided except where a real emergency exists. Even in the case of shore waters showing a satisfactory analysis, they will be found to contain a high quantity of air and other gases which also cause very serious corrosion in boilers. If more information were available as to the quality of these waters and more restrictions placed on their use, this source of scale and corrosion in Naval boilers could be materially reduced if not entirely eliminated. In the case of necessity of using shore waters, it should be required that they be distilled before use for feed purposes in order to insure the riddance of impurities and eliminate this possible source of boiler deterioration. The practice of obtaining water for boiler use from shore should certainly be discouraged instead of given encouragement, as is the case at present. A much higher quality of water can be obtained by distilling by effi-

cient operation, and even though the cost will almost invariably be more than for water obtained from shore, a saving will result in the long run by a reduction in deterioration and cost of repairs. Of course if a ship is not able to operate the evaporator plant efficiently, and obtain high quality water, then the use of shore water must be resorted to, as a choice of the lesser of two evils. A good many engineers claim they are unable to distill as good water as they can get on shore, and it is this lack of proper knowledge of the matter which is responsible for the use of large quantities of shore water on board ship. It is so much easier to obtain water from shore than to carefully distill good water, so why bother with distilling. This attitude is taken because they are not aware that they can distill a higher quality of water and due to the fact that the engineering competition rules give encouragement to this practice.

It has never been clearly brought out that boilers require a higher quality water than is needed for other fresh water purposes on board ship. This has resulted in some confusion and a divergence in practices by the engineering personnel. Water that is entirely satisfactory for drinking is not necessarily satisfactory for boiler use, in fact it may be entirely unsuitable. Of course, in the case of certain waters obtained from shore in some ports where experience has shown that these waters are bad for boilers, this fact is known. This is only known though in those cases where the water has proven to be particularly objectionable. In distilled water there is also a variation in quality and while water of two to five grains of chlorine per gallon is perfectly satisfactory for drinking and other purposes, this is not a satisfactory water for boilers. If pure distilled water can be obtained for boilers, then that is the quality of water that should be used in boilers.

Ships of the Navy are all designed to distill sufficient water for all its purposes including a fairly liberal allowance for make up feed, so that there is no real necessity, with favorable

operation, to obtain water from shore particularly for boiler use. The present rated capacity of these plants is based on the distilling of water of 2.0 grains of chlorine per gallon which is not considered of sufficiently high quality, and this capacity should be based on water of less than one grain per gallon. Water of two grains per gallon is not a good quality water for boilers and the rating should be made on a basis of satisfactory distilled water for boiler use. In the future design of evaporator plants a higher capacity should be allowed in order to insure a large margin of reserve water output for boiler purposes in order to reduce the necessity of obtaining water from shore. This would also afford reserve for quick renewal of water in case of high salinity due to a leaky condenser or other unavoidable cause.

All the engineering personnel on board ship are acquainted with the innumerable ways in which sea water can find its way into the boilers and are continually on their guard against them. The present practices prescribed of testing and checking to detect leaks into the feed system, if strictly followed, will result in quick detection of leaks and a minimum salting through these causes. By proper design, a good many sources of salt water entering fresh water have been eliminated or made fool-proof. The condensers, distillers, feed tanks, reserve tanks, and evaporated plant cannot be eliminated in this way and they require constant and vigilant attention and observation to keep them eliminated as a source of salt water entry into the feed system. It can be readily seen that if the engineering plant is kept free from salt leaks into the feed water system, then the only remaining source of salt water will be from the make up feed supplied by the evaporator plant. It is known that, in the operation of an evaporator plant, water of varying quality will be made and when the plant is being well operated that very good water will be distilled. It is also known that even with the most skilled operator there will be some interval of time during some watches when brackish

water will be made, particularly occurring when operating conditions are fluctuating. In the case of less expert operators, these periods of making brackish water will occur more frequently and to a greater degree depending on the efficiency of the operator. In other words, the human element is a large factor in distilling water and cannot be eliminated; therefore, this operation must be given considerable care and attention to obtain the desired results.

On board the U.S.S. *Idaho*, after commissioning, in a desire to keep the boiler water in very good condition, to reduce the amount of make up feed used, to lead to economy and minimum corrosion, every effort was given to following closely the best practices. The importance of a good start was fully realized. Every source of leakage into feed system was eliminated, and the system was maintained tight. As time went on it was decided the salinity of the boilers was still too high and that the make up feed water in reserve feed tanks was responsible for this. The tests of reserve feed tanks, over a period of several days, would show a variable degree of salinity varying from 0.5 grain of chlorine per gallon to five or more grains per gallon. This variation in the quality of reserve feed water was soon traced to the variable operation of the evaporator plant. The practice of distilling into the reserve feed tanks direct, when water was needed for make up feed, was used, this being the general practice on board all naval ships. It was then realized that insufficient attention was being paid to this matter and that as a result a variable quality of water was being supplied for boiler feed. It was then recalled that on several occasions poor operation of the evaporator plant had resulted in the salting of reserve feed tanks, and the bad water had to be either pumped overboard or used to wash out boilers being overhauled. When it is realized that these tanks are large, about 20,000 gallons capacity, and that a very short interval of poor operation of the evaporators will spoil a tank almost full of good quality water, the importance of eliminating this source of salt into the boiler

feed became very evident. It was then decided not to distill water into the reserve feed tanks, but to distill all water into the ship's tanks. After filling a ship's tank, the water was tested to determine its quality. The ship's tanks were smaller and realizing the fact that as high grade water was not needed for general ship's use as was needed for boilers, made this the logical decision. In case of salting of water a smaller quantity would be lost and in case the water was not of the best quality, then it could be used for ship's purposes. When it was desired to have water for boiler feed, the ship's tank showing the best grade water by test was transferred to the reserve feed tanks. This practice soon eliminated the fluctuation of the salinity of the reserve feed tanks and produced a very marked reduction in the salinity of the boilers, only the highest quality of water being transferred for make up feed purposes.

The arrangement of the fresh water system on the *Idaho* made it necessary to make the transfer at a special time generally after ten at night in order to prevent interference with the ship's water supply. This method of operation, although inconvenient, soon became routine and once for all eliminated this source of salt water from the feed water. This practice of testing each ship tank as soon as it was filled also furnished a closer check on the watch standers in the evaporator room. While the operators in the evaporator room test the quality of the water they are distilling, this testing is only done at intervals so that it is never an exact determination of their output for the watch. It is only human nature that in case an operator does make a small quantity of brackish water by poor operation, he will attempt to cover it up, realizing that it may not materially affect the total salinity. Unless it materially affects the quality of water, it will probably not be noticed, and even if it is noticed, it will be hard to place the responsibility on any particular watch stander. This all accounts for the varying quality of water made from day to day. In a plant making about 20,000 gallons of water per day, an analysis of

several lots with good evaporator operation, will show a variation from less than 0.5 grain of chlorine per gallon to five grains per gallon or more. This indicates that during some interval of time, depending on its analysis above 0.5 grain per gallon, that brackish water was being distilled. It will be found that in case of an investigation of this variation that all operators will claim that they are sure they made high quality water. When they find though that the quality of the water being made is tested carefully by an outsider, and watched by engineer officers, there will result a large improvement in the operation of the evaporator plant. When they know that only water of less than 0.7 grain of chlorine per gallon will be sent below for make up feed water, then they will all strive for this quality water.

It might be claimed that small tanks are provided in the evaporator room which makes it feasible to test water being distilled, and that this should be done. While this could be done, it is not practicable for so many tanks are made per hour, that it is not possible to make the evaporator men do this testing. In order to be sure that it is being done, and each tank of water satisfactory, it would be necessary to have a responsible man, not connected with the evaporator operation, conduct these tests. It must be realized that the operator is very much interested in his output. If the decision regarding the quality of water is left entirely in his hands, he will have a tendency to make a decision favorable to his output particularly when only small variations in quality are involved. With small tanks it is also considerable additional work to be changing the discharge of tanks from one place to another in case of variations of quality. If these tanks were large enough to hold the complete output of a watch and this water tested after the watch of each operator, this would furnish a complete check on the output of each watch stander, and would be an ideal arrangement. This design would require two tanks of sufficient capacity to hold the maximum output for the four-hour watch. When the water had been tested and its quality

determined, its use could be quickly decided, whether for make up feed, ships' tanks, or to be re-distilled. This would be the ideal arrangement for the evaporator plant design.

After giving this problem some attention, it was found on the *Idaho*, that very high quality water could be distilled, and all that was needed was to require this quality of water from the evaporator plant. The only reason why the best quality of water had not been generally obtained before was because it was not being demanded and no steps taken to determine its exact quality. Of course, it is not possible to continually obtain the best quality of water and variations in quality must be expected. Quite frequently a tank of water will show from two to five grains of chlorine per gallon or more. While this water can be used for general ship's uses, and is in every way satisfactory, it could not be used for extra feed without running up the salinity in the boilers.

The practice of distilling water direct to reserve feed tanks supplying a variable quality of water has been responsible for high salinity in boilers and bad salting of reserve feed tanks. This results in the encouragement to obtain shore water to keep down salinity, and also to replace shortage caused by salting of the reserve feed tanks. If this source of salt water is eliminated, will any ship need to obtain shore water, except in real emergencies?

The result of this method of obtaining feed water on the *Idaho* led to a marked reduction in salinity of the boilers. While before adopting this practice, it was difficult to maintain water in boilers at 100 grains of chlorine per gallon with a tight system, after eliminating this source of salt water, the salinity in the boilers was easily maintained at 25 grains per gallon and less. This reduction in salinity leads to better economy. Less feed water was required, due to less renewal of water in the boilers, and eliminating any necessity for blowing down. This reduction in feed water was reflected in the general efficiency of the entire plant. Less feed water being required, there was a reduction in the evaporator plant

output, reducing the strain on this plant to furnish water, and affording a larger available reserve capacity.

It is considered that too much latitude is allowed personnel in the present prescribed practices, in regard to the quantity of chlorine per gallon allowed in water in boilers. There are no restrictions or specific instructions covering this matter, and as a result there is a large divergence in practice. Some ships are able to maintain their boiler water at 25 grains per gallon or less, others are maintaining 50 grains per gallon, others 100 grains per gallon, 150 grains per gallon, etc. If some ships can carry boiler water over long periods at 25 grains per gallon, why cannot all other ships be made to follow, for it must show that their practices are not as good as on the ships carrying higher quality of boiler water. If more attention was given to this matter, by giving more publicity to the results being obtained on board ships, and by thorough inspections along this line to see why there are such differences in conditions, then undoubtedly all ships would realize the importance of low salinity and get busy and bring about this result. This would lead to a general improvement on board all ships.

In case of a leaky condenser, or other cause of leakage into the feed system, the salinity in boilers mounts very rapidly until the leak is eliminated. In these cases a good many ships do not take sufficiently prompt steps to renew this water. While low salinity may have been maintained in the boilers for a long period, when a casualty results in high boiler salinity, this salinity is allowed to remain high without prompt steps to renew this water, after correcting the source of leakage. Considerable corrosion in the boilers can be traced back to these periods when high salinity was maintained. If the engineering personnel would only realize that every day that high salinity is allowed to exist in boilers is materially increasing deterioration, more prompt action would be taken. It is certainly a matter that demands more urgent action than is at present given on board a good many ships. The removal of this high

salinity water should be considered just as important an emergency as the prompt correction of the leak responsible for this condition.

In reviewing the early histories of boilers of naval ships, it will be found that a good many ships' boilers are started on the downward path of deterioration by the shipyards after installation, and during the time they are in their charge, before delivery to the Government. This is due, of course, to lack of knowledge, a certain amount of indifference, and their not being held to sufficiently strict requirements in this respect. In the case of one battleship, careless practices led to a serious condition and a large repair and overhaul bill, shortly after being delivered to the Government. In the case of another battleship, the contractors used the boilers for a few months before being delivered to the Government, and they used filtered river water. Inspection of these boilers, shortly before being delivered, showed that they contained considerable mud, and that the water had not been properly filtered.

The contractors were required to clean out this mud, but the work was done in only a half-hearted way, so after commissioning, it was necessary to thoroughly clean the mud out of these boilers with the ship's force. This work required a period of about four months. Had the contractors been required to use a higher quality water, this would never have happened. Of course, in this instance, no evident material damage was done to the boilers, but it can be seen that serious consequences might have resulted. It certainly should not be allowed to take new boilers, the most vital part of the engineering plant, and allow them to get in a state that requires considerable overhaul immediately after delivery to the Government. The very strictest rules should be prescribed and enforced to prevent any possibility of such results.

There is another source of error that may exist in the care of boiler water, and that is in the analysis of the water. The instructions regarding the preparation of chemicals and the methods for testing are supplied to all ships, and if strictly

followed accurate results will be obtained. From past experiences with personnel in this respect, it is realized that there is carelessness, and that a good many ships do not get the accurate results that should be obtained. It is important to have the chemicals properly prepared for making the tests, otherwise inaccurate results will be obtained, even though the tests are properly made. This is of considerable importance for getting efficient and intelligent results with boiler operation and should be given close attention.

In order to enforce the best practices, an efficient inspection system should be established to visit all ships frequently. This inspection should check up the chemicals being used for testing, the accuracy of the tests made, the condition of the boilers and the salinity of the boilers. This inspection should also determine whether or not low salinity is being maintained and that the best practices are being followed. By publishing the results of the inspection, and the care being given to this matter by each ship, it could be compared with other ships and would lead to more attention being paid to this important matter. It would result in a more intelligent and thorough application of the best practices, improvement in practices and a reduction in boiler deterioration and repairs.

This inspection force could be organized under the Fleet Engineer and consist of a board of officers from several ships. This inspection would afford, in addition to checking up conditions on board ship, an opportunity of readily spreading among the personnel of all ships the best practices existing on the different ships in the care of water for boilers, and add to the general interest of this problem.

At the present time the proper care of boilers is a very important matter when it is considered that a large number of new vessels have been added to the Navy, and a large number is still to be added in the near future. When it is further realized that the engineering personnel of a good many of these ships is inexperienced, this problem becomes of addi-

BOIL

DATE	CONDENSERS.						FEED TANKS.											
	MAIN		DYNAMO		AUX.		MAIN				DYNAMO				#1			
	PORT	STBD.	PORT	STBD.	PORT	STBD.	PORT		STBD.		PORT		STBD.					
	SL.	SL.	SL.	SL.	SL.	SL.	SL.	AL.	SL.	AL.	SL.	AL.	SL.	AL.	SL.	AL.	SL.	

BOILER WATER TEST RECORD.

BOILERS.																		RESERV			
#1		#2		#3		#4		#5		#6		#7		#8		#9		#1		#2	
Sl.	Al.	Sl.	Al.	Sl.	Al.	Sl.	Al.	Sl.	Al.	Sl.	Al.	Sl.	Al.	Sl.	Al.	Sl.	Al.	Sl.	Al.	Sl.	

Sl = Salinity - Grs. p
Al = Alkilinity - %
Enter "E" when Boil
Enter "L" when Boil
Enter "F" when Boil
Enter "#" when Boil
These Entries to be
the Day Change

FIG. 1

BOILER WATER TEST RECORD.

BOILERS.																				RE	
#1		#2		#3		#4		#5		#6		#7		#8		#9		#1	
SL.	AL.	SL.	AL.	SL.	AL.	SL.	AL.	SL.	AL.	SL.	AL.	SL.	AL.	SL.	AL.	SL.	AL.	SL.	

SL = SALINITY -
AL = ALKILINITY
ENTER "E" WHE
ENTER "L" WHE
ENTER "F" WHE
ENTER "*" WHE
THESE ENTRIES
THE DAY CH

FIG. 1

	Reserve Feed Tanks								Boiler Compound Used		Remarks
	#1		#2		#3		#4				
Al.	Sl.	Al.	Sl.	Al.	Sl.	Al.	Sl.	Al.	Lbs.	Place	

...linity - Grs. per Gal.

...kilinity - %.

"E" when Boiler Empty.

"L" when Boiler Placed at Steam Level.

"F" when Boiler Filled.

"*" when Boiler Compound Added to Boiler Direct.

Entries to be Made Above "Sl" Entry for Day Change Occurs.

tional importance. Every effort should be made to educate the personnel to take proper care of these boilers and force them to realize their responsibility in this matter.

It has been found that in the care of boiler water, that the keeping of a "Boiler Water Test Record" book has been of great value, and forms a very valuable part of the history of boilers. This book shows, very quickly, and in an instructive form, all data of the tests of water in boilers and the different parts of the feed system, such as condensers, feed tanks, reserve feed tanks. This test record book is prepared as shown in Figure No. 1. In the preparation of this book, instead of placing the headings on the top of the page, place them on the left edge of the first page, and by cutting the edge of all the remaining pages in the book, so that the heading will show when the pages are turned over, this one heading will suffice for the book. This will allow for a continual record of the test of each part of the plant and places the data in a form that is useful in analyzing the results. A leaky condenser will affect the boiler salinity and this effect is shown at once in this record. The use of a similar boiler water test record book is highly recommended for all ships and when once adopted, it will soon be found to be of considerable value when used in connection with boiler inspection in studying the condition of the watersides and boiler deterioration. While all ships generally obtain all this data in their daily records, it is not placed in a useful form. This furnishes a continual and complete record of tests and makes it an important part of the history of boilers, condensers, and other parts of the feed system.

In reviewing this subject, in conclusion, it is considered that the methods available for improvement in quality of boiler water on board ship can be summarized as follows:

1. Elimination of salt in make up feed distilled.

2. Restriction or elimination of use of shore water for boiler purposes.

3. Distillation of shore water in case desired for boiler purposes.

4. Restriction of salinity content allowed in boiler water.

5. Education of personnel to follow best practices.

6. Enforcement of best practices on all ships by periodic and thorough inspections.

7. Stimulation of interest, by publication of results of practices on board ship.

NOTES.

FLOW OF WATER THROUGH CONDENSER TUBES.*

By William L. De Baufre, Lincoln, Neb.
and
Milton C. Stuart, Annapolis, Md.
Members of American Society of Mechanical Engineers.

In this paper the authors give particulars regarding an extended series of tests recently conducted at the United States Naval Engineering Experiment Station at Annapolis, Md., to determine the friction loss of water flowing through ⅝-inch No. 18 gage standard condenser tubes. The investigations covered variable velocities, water temperatures, and tube lengths, as well as the effect of both fresh and salt water. The results obtained are presented in tabular form and from them a general formula has been derived which gives the total drop in pressure due to entrance and exit losses and to frictional resistance within the tube.

There was conducted recently at the U. S. Naval Engineering Experiment Station, Annapolis, Md., an investigation upon the friction loss of water flowing through ⅝-inch standard condenser tubes of No. 18 gage, 0.522 inch in internal diameter. The principal variables were velocity and temperature of water, and tube length. The investigation also covered the variation in friction loss with clean tubes and tubes as received, and the effect of fresh and salt water. The computations were made in such a manner that losses at the tube ends and along the tube could be separated, the results being expressed in a general formula.

2. A number of sections of 3-inch iron pipe were asesmbled with flanges as indicated in Fig. 1. A corresponding number of lengths of ⅝-inch condenser tubing were obtained, and two brass blank flanges prepared to serve as tube sheets. These tube sheets could be inserted between any two pairs of flanges, thus enabling tube lengths of approximately 5, 8, 11, 14, 17 and 20 feet to be tested. The tubes were held in place by ordinary screwed glands in the tube sheets, and packed with cotton held between two fiber washers as shown in Fig. 2. They were supported by sheet-iron disks between the intermediate pairs of flanges.

3. Referring to Fig. 1, water from the storage tank S was pumped by the motor-driven centrifugal pump P through the heater H to the condenser tube indicated by the dotted lines. From the condenser tube the water was discharged into the two tanks W, where it was weighed before being discharged into the storage tank S below. The centrifugal pump P ran at constant speed, the rate of flow of water being regulated by valve V. The temperature of the water as indicated on thermometers T_1 and T_2, was regulated by the valve U admitting steam to the heater H.

* Paper presented at the Annual Meeting, December, 1919, of the American Society of Mechanical Engineers, 29 West 39th Street, New York. Reprinted by permission of and cuts furnished by A. S. M. E.

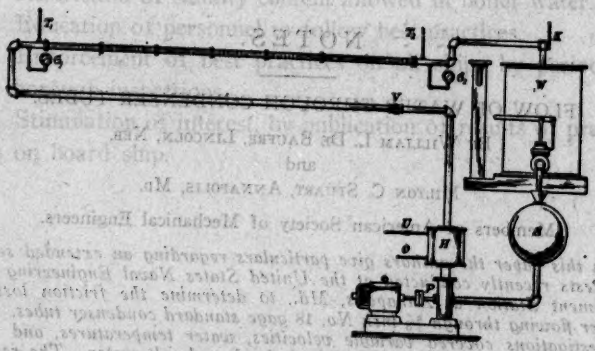

FIG. 1.—DIAGRAM OF TEST APPARATUS.

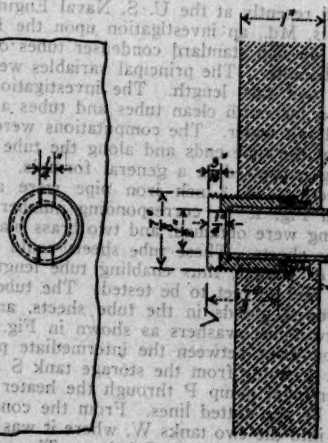

FIG. 2.—METHOD OF SECURING CONDENSER TUBE IN TUBE SHEET.

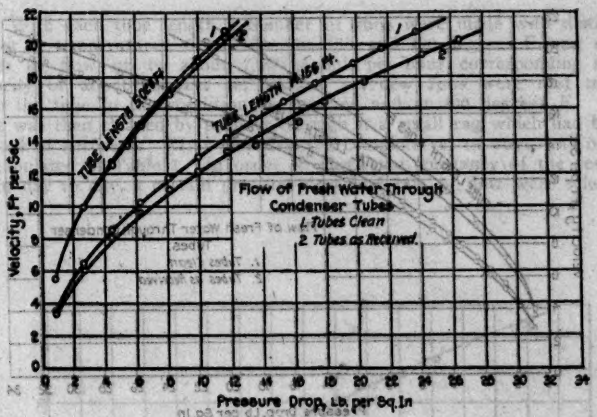
Velocity, Ft. per Sec.
Pressure Drop, Lb. per Sq. In.
TUBE LENGTH 5.04 FT.
TUBE LENGTH 11.156 FT.
Flow of Fresh Water Through Condenser Tubes
1. Tubes Clean
2. Tubes as Received.

FIG. 3.—CURVES SHOWING THE EFFECT OF CLEANING TUBES.

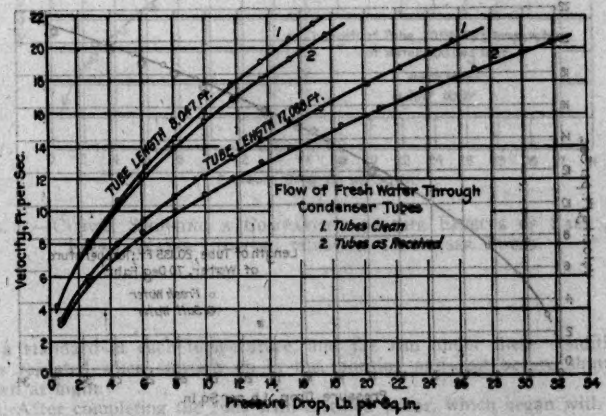
Velocity, Ft. per Sec.
Pressure Drop, Lb. per Sq. In.
TUBE LENGTH 5.04 FT.
TUBE LENGTH 11.156 FT.
Flow of Fresh Water Through Condenser Tubes
1. Tubes Clean
2. Tubes as Received

FIG. 4.—CURVES SHOWING THE EFFECT OF CLEANING TUBES.

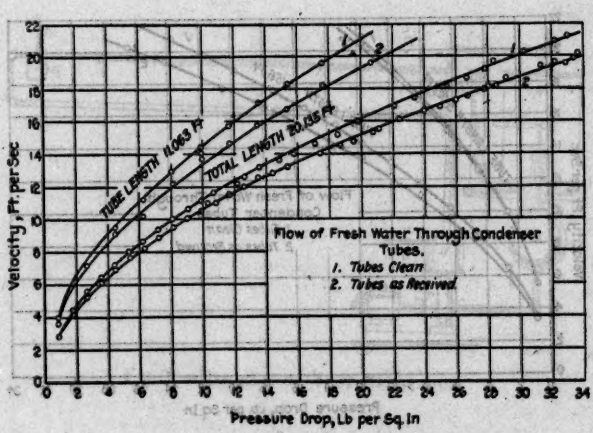
Velocity, Ft. per Sec
Pressure Drop, Lb per Sq In
TUBE LENGTH 17.063 FT.
TOTAL LENGTH 20.135 FT.
Flow of Fresh Water Through Condenser Tubes.
1. Tubes Clean
2. Tubes as Received

FIG. 5.—CURVES SHOWING THE EFFECT OF CLEANING TUBES.

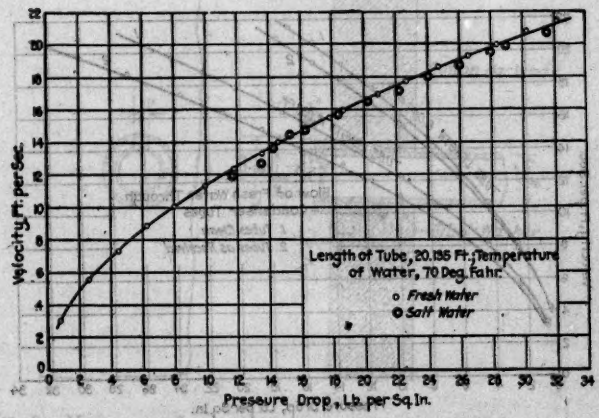
Velocity, Ft. per Sec.
Pressure Drop, Lb per Sq In.
Length of Tube, 20.135 Ft.; Temperature of Water, 70 Deg. Fahr.
o Fresh Water
⊙ Salt Water

FIG. 6.—CURVES SHOWING A COMPARISON OF THE EFFECTS OF SALT AND FRESH WATER FLOWING THROUGH CONDENSER TUBES.

The condensed steam was discharged through O into a trap not shown. Gages G_1 and G_2 served to measure the pressures before and after the tube.

4. With each tube length a number of runs were made with distilled water at temperatures of 85, 100, 130, 160, and 190 degrees F. and with rates of flow up to about 7,500 pounds per hour, corresponding to a velocity of about 22 feet per second. A few runs were first made with the tube in the condition as received and at 100 degrees F. The tube was then cleaned by pushing through it a small rag which had been soaked in kerosene. After cleaning, the complete set of runs was made as tabulated in Table 1. In order to check the constancy of the results from day to day, a certain pressure drop and rate of flow were selected

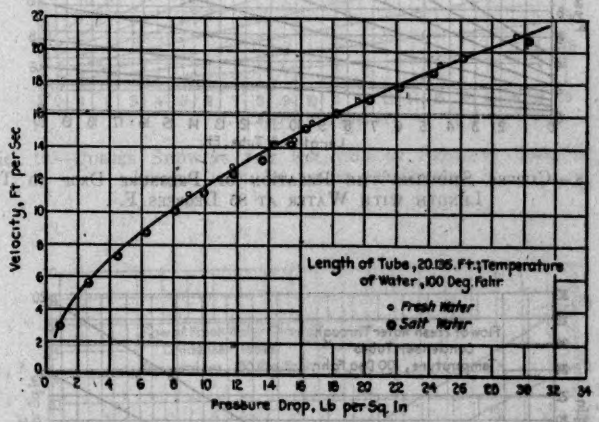
Velocity, Ft per Sec
Pressure Drop, Lb per Sq In
Length of Tube, 20.136 Ft.; Temperature of Water, 100 Deg. Fahr
o Fresh Water
⊕ Salt Water

FIG. 7.—CURVES SHOWING A COMPARISON OF THE EFFECTS OF SALT AND FRESH WATER FLOWING THROUGH CONDENSER TUBES.

as a standard at each temperature, and the run under these conditions was repeated when starting up in the morning and just before shutting down at night.

5. After completing the runs with distilled water, which began with the longest tube and ended with the shortest one, the longest tube was again installed in order to make runs with salt water and compare the results with the data obtained for fresh water. A few runs were repeated with fresh water and then duplicated with salt water at both 70 and 100 degrees F. The salt water was prepared by adding to the fresh water one thirty-second of its weight of salt, in order to approximate the normal density of sea water.

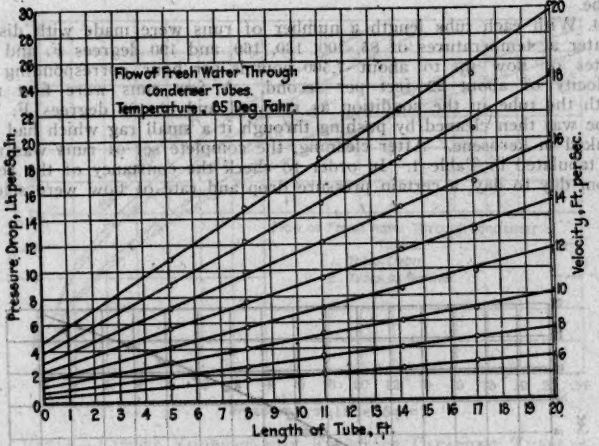
Flow of Fresh Water Through
Condenser Tubes.
Temperature, 85 Deg. Fahr.
Pressure Drop, Lb. per Sq. In.
Velocity, Ft. per Sec.
Length of Tube, Ft.

FIG. 8.—CURVES SHOWING THE RELATION OF PRESSURE DROP TO TUBE LENGTH WITH WATER AT 85 DEGREES F.

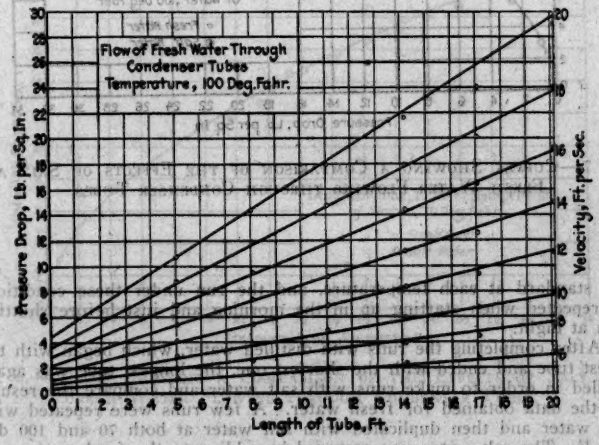
Flow of Fresh Water Through
Condenser Tubes
Temperature, 100 Deg. Fahr.
Pressure Drop, Lb. per Sq. In.
Velocity, Ft. per Sec.
Length of Tube, Ft.

FIG. 9.—CURVES SHOWING THE RELATION OF PRESSURE DROP TO TUBE LENGTH WITH WATER AT 100 DEGREES F.

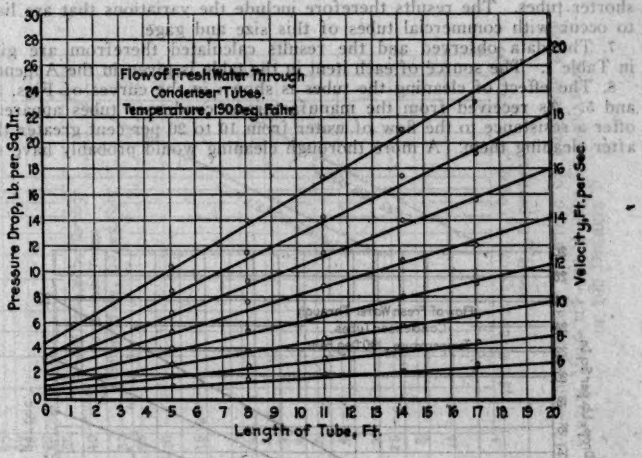
Flow of Fresh Water Through Condenser Tubes. Temperature, 130 Deg. Fahr.
Pressure Drop, Lb. per Sq. In.
Velocity, Ft. per Sec.
Length of Tube, Ft.

FIG. 10.—CURVES SHOWING THE RELATION OF PRESSURE DROP TO TUBE LENGTH WITH WATER AT 130 DEGREES F.

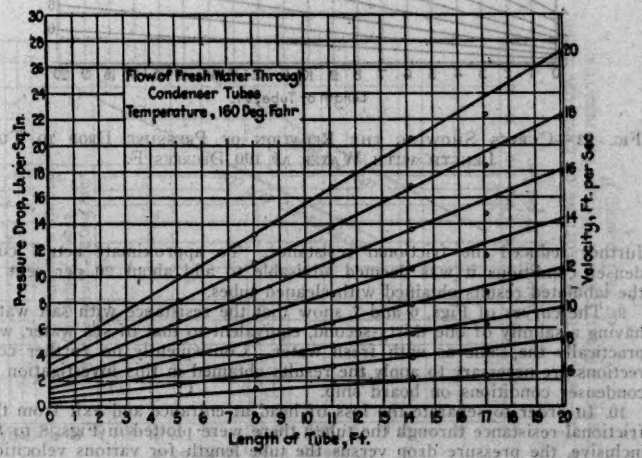
Flow of Fresh Water Through Condenser Tubes Temperature, 160 Deg. Fahr.
Pressure Drop, Lb. per Sq. In.
Velocity, Ft. per Sec.
Length of Tube, Ft.

FIG. 11.—CURVES SHOWING THE RELATION OF PRESSURE DROP TO TUBE LENGTH WITH WATER AT 160 DEGREES F.

6. For the several tube lengths it was decided to use entirely different tubes rather than to take one long tube and cut off parts to obtain the shorter tubes. The results therefore include the variations that are liable to occur with commercial tubes of this size and gage.

7. The data observed and the results calculated therefrom are given in Table 1. The source of each item in the table is given in the Appendix.

8. The effect of cleaning the tubes is shown in the curves of Figs. 3, 4 and 5. As received from the manufacturers, condenser tubes apparently offer a resistance to the flow of water from 10 to 20 per cent greater than after cleaning them. A more thorough cleaning would probably have still

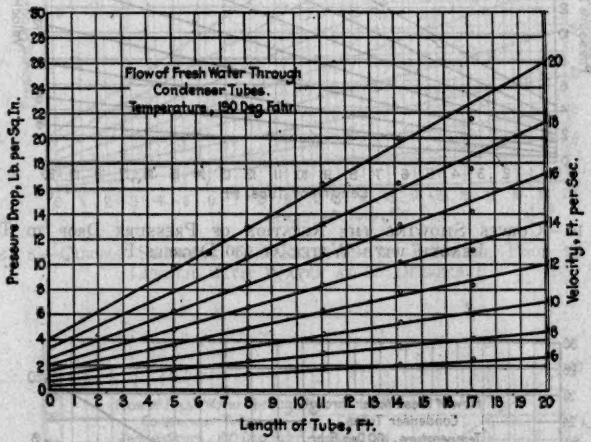
Flow of Fresh Water Through Condenser Tubes. Temperature, 190 Deg. Fahr.
Pressure Drop, Lb. per Sq. In.
Velocity, Ft. per Sec.
Length of Tube, Ft.

FIG. 12.—CURVES SHOWING THE RELATION OF PRESSURE DROP TO TUBE LENGTH WITH WATER AT 190 DEGREES F.

further reduced the frictional resistance. To approximate actual condenser conditions it was deemed advisable to add about 20 per cent to the tabulated results obtained with cleaned tubes.

9. The curves of Figs. 6 and 7 show that the resistance with salt water having a salinity of one thirty-second, equivalent to that of sea water, was practically the same as with fresh water. Consequently no further corrections are necessary to apply the results obtained in this investigation to condenser conditions on board ship.

10. In order to separate the loss of head at entrance and exit from the frictional resistance through the tubes, there were plotted in Figs. 8 to 12, inclusive, the pressure drop versus the tube length for various velocities. The points plotted in these figures were not taken directly from Table 1, but from faired curves (not included in this paper) of all runs made.

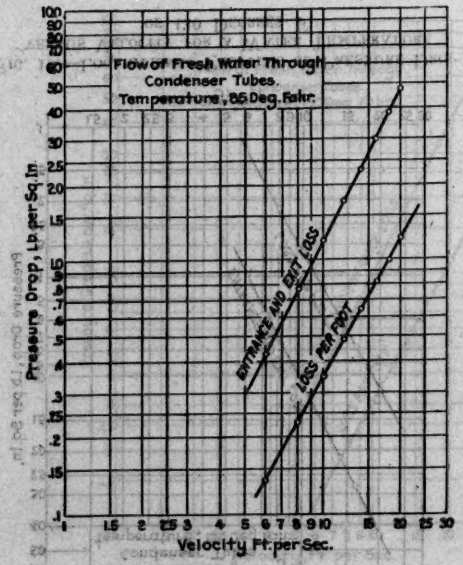
Flow of Fresh Water Through
Condenser Tubes.
Temperature, 85 Deg. Fahr.
ENTRANCE AND EXIT LOSS
LOSS PER FOOT
Pressure Drop, Lb per Sq. In.
Velocity Ft. per Sec.

FIG. 13.—LOGARITHMIC PLOTTING OF PRESSURE DROP VERSUS VELOCITY FOR A WATER TEMPERATURE OF 85 DEGREES F.

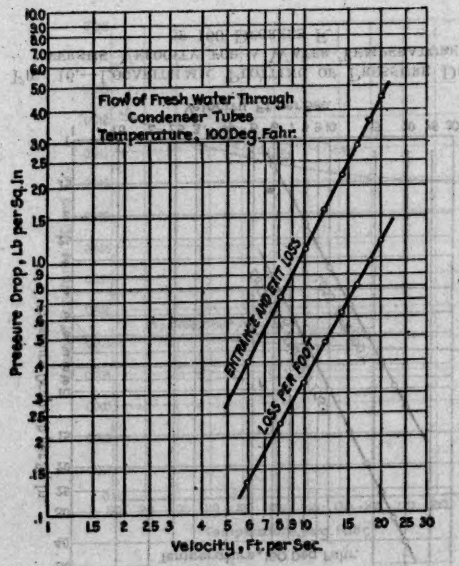
Flow of Fresh Water Through
Condenser Tubes
Temperature, 100 Deg. Fahr.
ENTRANCE AND EXIT LOSS
LOSS PER FOOT
Pressure Drop, Lb per Sq. In.
Velocity, Ft. per Sec.

FIG. 14.—LOGARITHMIC PLOTTING OF PRESSURE DROP VERSUS VELOCITY FOR WATER TEMPERATURE OF 100 DEGREES F.

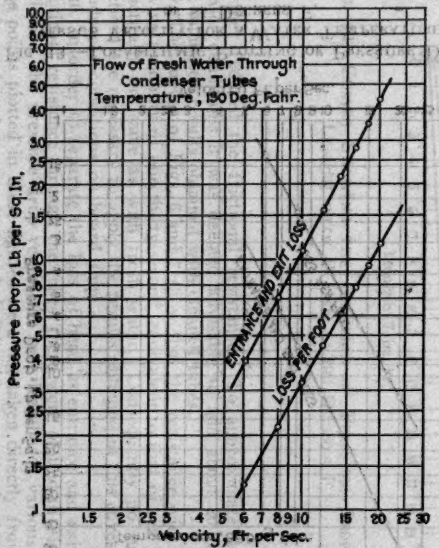
Flow of Fresh Water Through
Condenser Tubes
Temperature, 130 Deg. Fahr.
ENTRANCE AND EXIT LOSS
LOSS PER FOOT
Pressure Drop, Lb. per Sq. In.
Velocity, Ft. per Sec.

FIG. 15.—LOGARITHMIC PLOTTING OF PRESSURE DROP VERSUS VELOCITY FOR A WATER TEMPERATURE OF 130 DEGREES F.

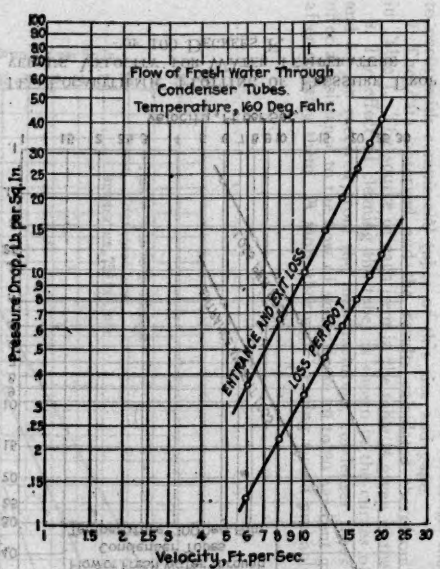
Flow of Fresh Water Through
Condenser Tubes
Temperature, 160 Deg. Fahr.
ENTRANCE AND EXIT LOSS
LOSS PER FOOT
Pressure Drop, Lb. per Sq. In.
Velocity, Ft. per Sec.

FIG. 16.—LOGARITHMIC PLOTTING OF PRESSURE DROP VERSUS VELOCITY FOR A WATER TEMPERATURE OF 160 DEGREES F.

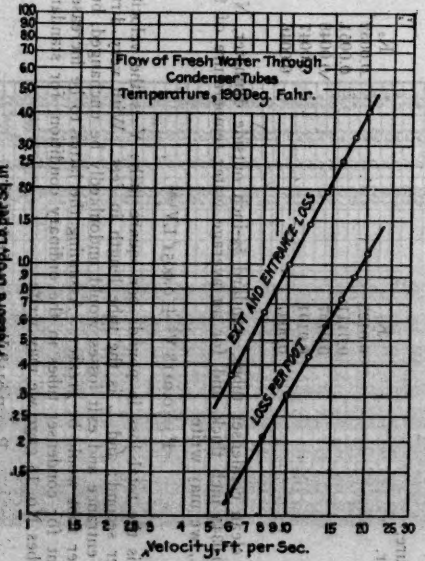
Flow of Fresh Water Through Condenser Tubes
Temperature, 190 Deg. Fahr.
EXIT AND ENTRANCE LOSS
LOSS PER FOOT
Pressure Drop, Lb. per Sq. In.
Velocity, Ft. per Sec.

FIG. 17.—LOGARITHMIC PLOTTING OF PRESSURE DROP VERSUS VELOCITY FOR A WATER TEMPERATURE OF 190 DEGREES F.

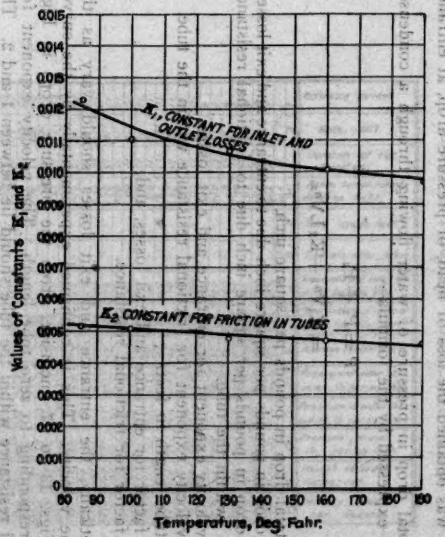
K_1, CONSTANT FOR INLET AND OUTLET LOSSES
K_2 CONSTANT FOR FRICTION IN TUBES
Values of Constants K_1 and K_2
Temperature, Deg. Fahr.

FIG. 18.—CURVES SHOWING THE RELATION OF CONSTANTS TO THE TEMPERATURE OF WATER.

By prolonging the straight lines for each velocity back to zero tube length there was obtained the loss of head, or pressure drop, at entrance and exit.

11. The total drop in pressure of water flowing through a condenser tube may be expressed by the formula—

$$P = P_1 + P_2$$
$$= K_1 V^{n_1} + K_2 L V^{n_2}$$

where P = total drop in pounds per square inch.
P_1 = drop in pounds per square inch due to entrance and exit losses.
P_2 = drop in pounds per square inch due to frictional resistance within the tube.
n_1 = velocity exponent for entrance and exit losses.
n_2 = velocity exponent for frictional resistance within the tube.
L = tube length in feet.
K_1 = factor for entrance and exit losses, and,
K_2 = factor for frictional resistance.

12. Theoretically, the entrance and exit losses should vary as the square of the velocity. This is confirmed experimentally by the curves plotted in Figs. 13 to 17, inclusive, from the pressure drops in Figs. 8 to 12, corresponding to zero tube length. The velocity exponent for the frictional resistance within the tube should lie between 1 and 2. The curves in Figs. 13 to 17 for the loss per foot of tube length were obtained by plotting the slopes of the straight lines drawn in Figs. 8 to 17. The exponent in all cases was found to be 1.83.

13. The following values of K_1 and K_2 were taken from the curves of Figs. 12 to 16, inclusive, and are plotted in Fig. 18.

Temperature, deg. Fahr.	K_1	K_2
85	0.0123	0.0052
100	0.0111	0.0051
130	0.0107	0.0048
160	0.0101	0.0047
190	0.0097	0.0046

14. For clean condenser tubes, standard ⅝-inch outside diameter, No. 18 gage, 0.0345 inch thick, and for an average water temperature of 90 degrees F., we may write

$$P = 0.0118\ V^2 + 0.0051\ LV^{1.83}$$

where P is the total loss in pounds per square inch; V is the velocity in feet per second; and L is the tube length in feet. With very dirty tubes the entrance and exit losses would undoubtedly be unchanged, but the loss per foot would be greater. Assuming the latter to be increased 20 per cent for condenser tubes in the ordinary condition, for standard ⅝-inch tubes No. 18 gage we may write

$$P = 0.0118\ V^2 + 0.0061\ LV^{1.83}$$

Expressing the resistance in feet head of water, we have

$$H = 0.0274\ V^2 + 0.0141\ LV^{1.83}$$

TABLE 1 RESULTS OF INVESTIGATION ON FRICTION LOSS OF WATER FLOWING THROUGH CONDENSER TUBES

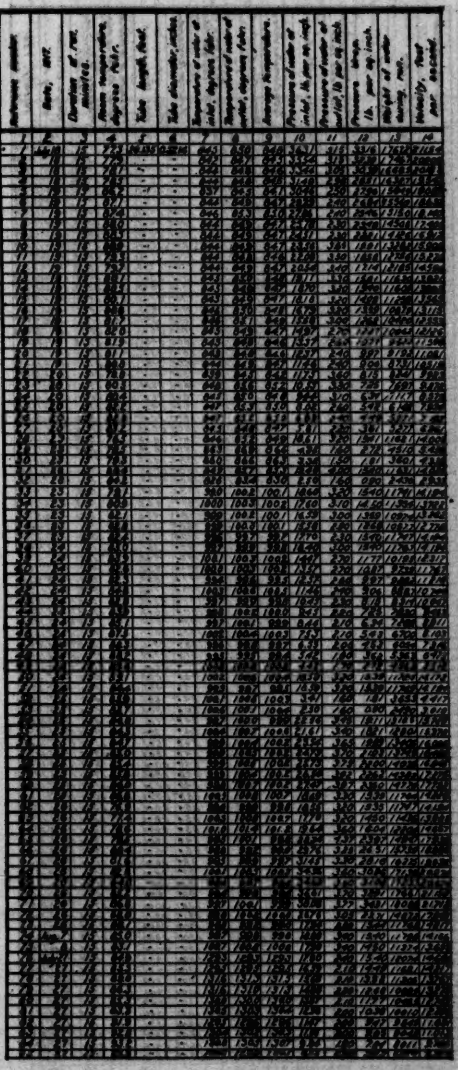

TABLE 1 (*Cont.*)

TABLE 1 (*Cont.*)

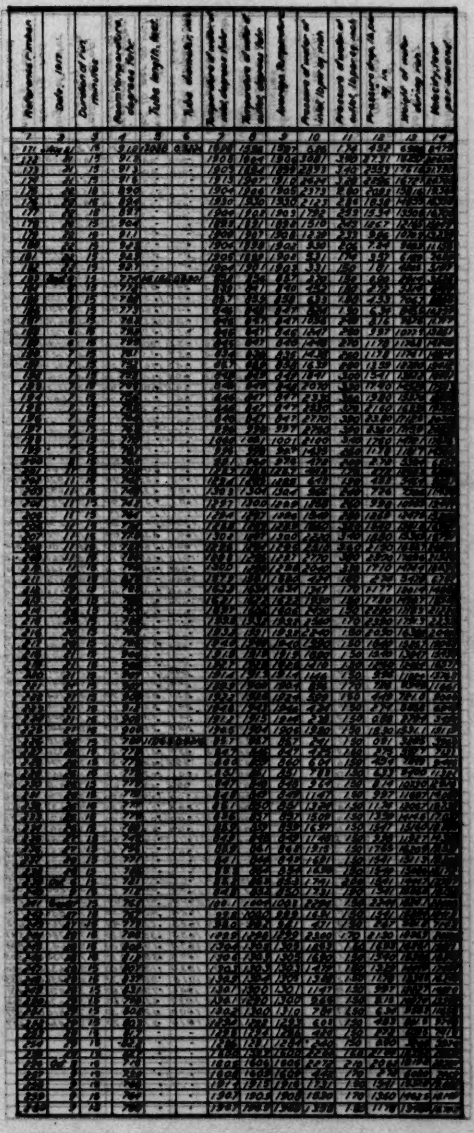

TABLE 1 (*Cont.*)

Reference number	Date, 1917	Duration of run, minutes	Room temperature, degrees Fahr.	Tube, length, feet	Tube, diameter, inches	Temperature of water inlet, degrees Fahr.	Temperature of water outlet, degrees Fahr.	Average temperature of water, degrees Fahr.	Pressure of water in inlet, lb. per sq. inch	Pressure of water at outlet, lb. per sq. inch	Pressure drop, lb. per sq. inch	Weight of water during run, lb.	Velocity, feet per second
1	2	3	4	5	6	7	8	9	10	11	12	13	14
261	Oct 9	15	776	11.083	0.366	190.7	191.0	190.8	8.66	1.79	6.87	1005.3	13.38
262	" 9	15	776	"	"	191.8	191.1	191.6	8.66	1.79	6.87	1010.6	13.44
263	" 9	15	776	"	"	191.9	191.8	191.9	8.66	1.79	6.87	1010.6	13.44
264	" 10	15	703	"	"	182.0	181.5	181.8	23.78	8.16	15.62	967.6	12.87
265	" 10	15	723	"	"	189.6	189.1	189.3	23.78	8.16	15.62	963.5	12.81
266	" 10	15	722	"	"	190.8	190.4	190.6	23.78	8.16	15.62	963.5	12.81
267	" 10	15	744	"	"	191.5	191.1	191.3	8.66	1.79	6.87	1010.6	13.44
268	" 10	15	744	"	"	191.5	191.1	191.3	8.66	1.79	6.87	1010.6	13.44
269	" 10	15	807	"	"	194.8	194.1	194.5	8.66	1.79	6.87	1010.6	13.44
270	" 10	15	807	"	"	194.8	194.1	194.5	8.66	1.79	6.87	1010.6	13.44
271	" 10	15	806	"	"	194.8	194.3	194.5	8.66	1.79	6.87	1010.6	13.44
272	" 10	15	806	"	"	194.8	194.3	194.5	8.66	1.79	6.87	1010.6	13.44
273	" 10	15	806	"	"	194.8	194.3	194.5	8.66	1.79	6.87	1010.6	13.44
274	" 10	15	806	"	"	194.8	194.3	194.5	8.66	1.79	6.87	1010.6	13.44
275	" 10	15	783	11.083	0.366	193.8	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
276	" 10	15	783	"	"	193.8	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
277	" 10	15	783	"	"	193.8	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
278	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
279	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
280	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
281	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
282	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
283	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
284	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
285	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
286	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
287	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
288	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
289	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
290	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
291	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
292	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
293	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
294	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
295	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
296	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
297	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
298	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
299	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
300	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
301	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
302	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
303	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
304	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
305	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
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307	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
308	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
309	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
310	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
311	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
312	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
313	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
314	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
315	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
316	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
317	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
318	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
319	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
320	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
321	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
322	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
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324	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
325	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
326	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44
327	" 10	15	781	"	"	193.7	193.4	193.6	11.57	1.80	9.77	1010.6	13.44

APPENDIX.

SOURCES OF ITEMS IN TABLE I.

ITEMS.

1. Reference number.
   For reference.
2. Dates.
   From records taken during run.
3. Duration of run, minutes.
   From records taken during run.
4. Room temperature, degrees Fahrenheit.
   From thermometer readings taken during run.
5. Tube length, feet.
   From measurement.
6. Tube diameter, inches.
   From measurement with micrometers, checked by weight of water contained.
7. Temperature of water inlet, degrees Fahrenheit.
   From thermometer readings taken during run corrected for calibration error.
8. Temperature of water outlet, degrees Fahrenheit.
   From thermometer readings taken during run corrected for calibration error.
9. Average temperature, degrees Fahrenheit.
   Average of inlet and outlet temperatures, Items 7 and 8.
10. Pressure of water at inlet, pounds per square inch.
    From gage and mercury-column readings corrected for calibration error.
11. Pressure of water at outlet, pounds per square inch.
    From gage and mercury-column readings corrected for calibration error.
12. Pressure drop, pounds per square inch.
    Item 10 minus Item 11.
13. Weight of water during run.
    From readings taken on calibrated scales.
14. Velocity of water, feet per second.
    Calculated from formula, $V = Q/A$, where Q is the quantity in cubic feet per second calculated from Item 13 and a density corresponding to the average temperature (Item 9), and A is the area in square feet corresponding to the diameter (Item 6).

HIGH-SPEED CRAFT DURING THE WAR.*

To be able to move quickly, apart from its utility, has always a charm, and the advantages of high speed in warfare are so evident that it is hardly necessary to emphasize them. A higher speed than that of the enemy gives the advantage of position, and also the power of accepting battle or not, so that full advantage may be taken of any superiority in armament, or if the enemy has the more powerful weapons it is possible to keep out of his way.

Speed, however, cannot be obtained without paying the price, and whether it is the rate at which one lives, or the speed of an express train or the speed of a vessel, in every case something has to be sacrificed in order to obtain what is desired. The amount of energy at our disposal is limited, and if we want to move quickly we must reduce the resistance to a minimum and utilize the maximum energy.

I have had the honor of being associated with Messrs. Yarrow during the whole of my career, and consequently my description in this paper of high-speed vessels is limited to what has been accomplished by this firm. I am quite sure that Sir Alfred Yarrow and Mr. Yarrow would be the very last to refuse a tribute to what has been done by other firms and individuals in the development of high-speed vessels.

One of the most renowned firms in this country is that of Messrs. Thornycroft, and the development of high-speed vessels owes much to the bold advances which have been successfully made by that firm from time to time, both in design and in taking advantage of the best available materials. Not only did they build many high-speed destroyers during the war, but they evolved a most successful high-speed coastal motor boat. The feats accomplished by these coastal motor boats are some of the most remarkable of the war. Any results which may have been obtained by them are, in the first instance, due to the skill and endurance of the officers and men of the Navy, but Messrs. Thornycroft deserve the highest praise for designing and building such wonderful craft.

Mention might also be made of the splendid work done in the early days by M. Normand, the great French designer, our friend and rival. As an instance of business chivalry it might be of interest to you to relate an incident which happened many years ago when rivals were struggling to be the first to obtain 30 knots, and in the end it came to a neck and neck race between the firms of Normand and Yarrow. The Yarrow boat was finished first, and on trial obtained the coveted speed of 30 knots. Concerning this performance the first congratulatory telegram received by Messrs. Yarrow was from M. Normand, and it is a telegram which we prize more than any other received at the time.

The title of the paper is "High-Speed Vessels during the War," and this enables me to draw attention to the splendid work done by passenger vessels, by high-speed steam yachts, by racing vessels of all kinds, and by hydroplanes, and it will be agreed that the lessons learned in some of these sporting vessels have advanced our knowledge in a most useful way.

In acknowledging the magnificent work which has been achieved by other builders of destroyers, the worker in science, and the pioneers in the various fields of industry, such as steelmakers, makers of turbines, and makers of auxiliary machinery, should not be forgotten, all of whom have contributed to making high-speeds possible.

In the design of a high-speed vessel it is necessary to make a careful study of the conditions under which she has to operate. A vessel for a

*Paper read before the Institution of Engineers and Shipbuilders in Scotland, April 20, by Mr. W. W. Marriner.

shallow river is quite a different proposition from a vessel operating at sea, and again a high-speed vessel which has to operate on a short radius of action is very different from one which has to operate at a great distance from its base. The governing principle, however, remains the same, namely, resistance must be reduced to a minimum and power must be obtained most efficiently.

Consider, first of all, the question of weight, which is a large factor in resistance. It may be divided into weight of hull, weight of machinery, weight of crew and armament, and weight of fuel. The weight of armament is fixed by the military authorities (especially those of the enemy) as it is useless to get into a position to strike a blow, unless on arrival the blow can be made effective.

The total weight is largely influenced by the all-important consideration of cost. Having determined the total displacement it is then possible to allocate what weight can be allowed for the hull and for the machinery respectively. If the vessel has only to make short raids and return immediately to her base, then it is obvious that the scantlings of the hull and the accommodation for the crew can be reduced, because the vessel can choose the weather in which she puts to sea, and the crew can put up with a fair amount of hardship for a short period.

In the late war the Germans made full use of this condition, and their destroyers used to put to sea with a minimum of fuel and outfit on board, with the consequence that in many cases they were able to outdistance British vessels which had naturally to be prepared for keeping the sea for an indefinite period in all kinds of weather.

Generally speaking, in the modern destroyer the distribution of weights is approximately as follows:—

	Per Cent.
Weight of hull	35
Weight of machinery	35
Weight of armament and stores	7
Weight of fuel	23

Having settled the weight which can be allowed for the hull, one can decide the dimensions of the vessel. For speed the important dimension is the length, for stability the important dimension is beam, and for strength the important dimension is depth. The length is, therefore, influenced by its relation to the beam and depth of the vessel, and it is here where the skill of the naval architect is shown by obtaining the very best compromise as regards the dimensions to be adopted, always keeping in view the special service for which the boat is to be employed. A vessel which has to be constantly employed with a fleet, and whose speed is determined by the senior officer of a larger ship, should be made stronger than the vessel, which will only have to act independently or with vessels of its own class, when the senior officer of the flotilla is himself experiencing the shocks caused by waves. Even in a comparatively calm sea there is a certain speed for every vessel where the synchronism of the waves with pitching might become serious.

Much can be done by arranging weights relatively to the buoyancy at any part, so as to reduce the bending moments, and present-day lightness in hull construction has been attained without any sacrifice of strength or decreasing the factor of safety, by a scientific study of the stresses involved and the best disposition of the material.

Having decided upon the dimensions of the vessel, the next important consideration is the form of the under-water hull, and in this connection tank experiments are valuable; in fact, essential. One can hardly believe

that twenty years ago it was necessary to go to Germany in order to have tank experiments carried out, although there were private tanks in this country which, however, were not available for public use. The form of the vessel is important not only from the point of view of reducing the resistance of the hull to a minimum, but also to ensure that the propeller works with the highest possible efficiency. From tank experiments it is possible to form a close estimate of the power required to drive the vessel, and for a given speed it is just as important not to over-estimate the power as it is not to underestimate it, because the weight of the machinery may be taken to be proportional to the power, and excessive weight of machinery means a load to be carried increasing also the weight of the hull, and thus doubly reducing the speed.

Possibly sufficient attention has not been paid to the underwater form from the point of view of obtaining the highest propeller efficiency, especially in vessels where the power is proportionately very large. It is well known that at high speeds the propeller is one of the most inefficient parts of the propelling installation. Models have been equipped with propellers driven by motors installed on the model and a measure taken of the power required by the motors. Naval architects in this country at one time would not place much reliance on results obtained in this way, but owing to the researches carried out by Mr. R. E. Froude at Haslar, and Mr. Baker at the William Froude National Tank, Teddington, means have been perfected by which accurate investigations can be made, and experience has proved that the methods adopted by them are sufficiently accurate to form a reliable guide. As long ago as 1903 Sir Alfred Yarrow advocated a flat-bottom stern for a destroyer in order to increase the propeller efficiency, and H.M.S. *Garry* may be looked upon as an early example of this type of stern for destroyers. The resistance of the hull is hardly affected as compared with the ordinary round-bottom stern, but the propeller efficiency is increased to a remarkable extent, a knot more speed being obtained for the same expenditure of power.

Having fixed on the weight of the hull and machinery, the designer is then in a position to say how much weight can be allowed for the armament and fuel on a vessel of certain displacement. The importance of weight on a modern destroyer may be illustrated by stating the fact that for every pound weight thrown overboard the vessel will be ½ inch further ahead each hour, or for every ton weight saved half a length of the ship will be gained in an hour's race.

The relationship which exists between the various parts is one of the most important factors in the success of a destroyer, not only from the point of view of speed, but also from the point of view of seaworthiness. Any Government Department is in an unfavorable position in this respect as compared with a private firm, because in the latter there is one head who is in a position to settle what he considers to be the best compromise. Sometimes the hull has to be cut down, and sometimes the machinery, and sometimes it is necessary to say that the armament is not suitable for that particular size of vessel. In firms who make their own designs it has been noticed that every now and again a design is arrived at which gives results far ahead of what has been before obtained. For instance, in 1894, Sir Alfred Yarrow constructed the *Sokol* for the Russian Government, which was a vessel 190 feet long by 18 feet 6 inches beam, with twin-screw engines of a total indicated horsepower of about 4,000 and Yarrow water-tube boilers. Her armament comprised one 75-mm. gun, three 47-mm. guns, and two deck torpedo tubes. She was the first vessel to attain a speed on trial of 30 knots, and this result was obtained principally by the adoption of water-tube boilers, and high-tensile steel in the construction of the hull. This special steel had a tensile

strength of from 37 tons to 44 tons per square inch, and an extension of not less than 15 per cent in 8 inches for plates 3-16 inch thick. Its application permitted a reduction in thickness of 12 per cent below that of the mild steel previously used, with such a saving in weight of hull that an extra knot was added to the speed. All the stanchions were made hollow, pressed steel was used wherever possible instead of cast metal, and for cabin fittings aluminum was adopted, all in order to save weight. She was built at the same time as other destroyers which had a difficulty in obtaining 26 knots.

The *Sokol* was gradually improved upon in future designs. Keeping as near as possible to the same relationship of weights and form, and taking advantage of every modern improvement, what was considered one of the very best vessels they ever built, namely, the *Mode*, was constructed in 1902 for the Swedish Government. She was 220 feet 3 inches long by 20 feet 6 inches beam, and had four Yarrow boilers, and twin-screw reciprocating engines of a total indicated horsepower of 6,400. The speed realized on the official trial was nearly 32.5 knots. For the size and power this was a remarkable result, especially as it was obtained when burning coal under the boilers.

The adoption of oil fuel was perhaps the next great step forward in the evolution of destroyers, and an enormous amount of credit is due to the Admiralty Experimental Station at Portsmouth, which brought oil-fuel burning apparatus to the perfection which is now known. One cannot help feeling that there must be some engineer officers whose names ought to be made public so that due honor might be paid them for their work in connection with one of the greatest improvements for war vessels, especially as it placed this country far ahead of the rest of the world for many years.

In 1910 the Admiralty, at Admiral Jellicoe's initiative, gave Messrs. Yarrow almost a free hand to do the best they could with a certain sum of money. The result was an outstanding design from which the *Firedrake*, *Lurcher*, and *Oak* were built. Their dimensions were 255 feet long by 25 feet 7 inches beam, and each was fitted with Parsons' impulse reaction turbines and three oil-burning Yarrow boilers. One of these vessels on the official trial of 8 hours duration obtained a speed or 35.354 knots.

The next important advance was in the direction of superheating. It had been long known that superheating offered great advantages both as regards economy and reliability, but in spite of many attempts it was not looked upon with great confidence for marine purposes. It was, however, so attractive that a special marine superheater was designed and fitted on one side only of the Yarrow boiler, by which arrangement the superheater is placed under control, and that without seriously interfering with the total output of the boiler (see Figs. 1 and 2). If too much superheat is being obtained it is possible to send a larger proportion of the gases up the side of the boiler where there is no superheater, and although there might be a trifling loss in efficiency, yet the total output of the boiler is not substantially reduced. If the superheat is not sufficient, then more of the furnace gases can be sent up the side of the boiler on which the superheater is placed. This control of the amount of superheat was found very valuable, especially in the experimental stage.

It may here be mentioned that with a properly-designed Yarrow boiler having the superheater on one side that higher efficiencies of the boiler plant have been obtained than with a boiler without a superheater, that is to say, more heat units have been transferred from the oil fuel to the steam produced. This result is rather remarkable, and points to the high efficiency of heat transference in a well-designed superheater.

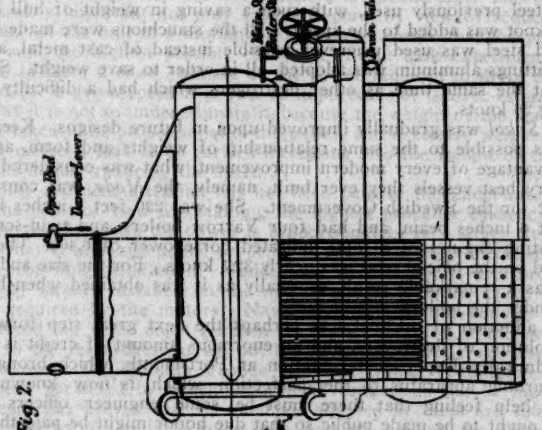


Fig. 2.

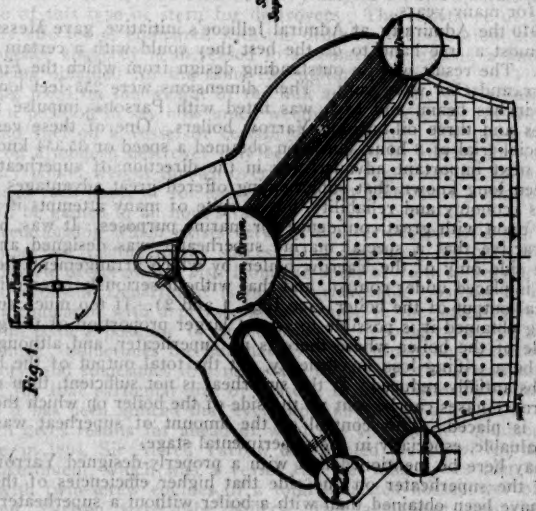


Fig. 1.

Owing to the adoption of superheating, in 1913 and 1914 we were again able to produce a destroyer which had a very high overall efficiency. Although superheating had been tried in previous vessels yet the most successful results were obtained in vessels of the *Miranda* Class in 1913 and in 1914, and it was on this type which Messrs. Yarrow concentrated during the war period.

These vessels showed plainly the advantage of superheating for war vessels apart from efficiency, namely, the ability to rapidly increase speed. It is necessary to exercise great care when increasing speed rapidly on account of the water which flows with the steam, and on account of the condensation which takes place in the engines and turbines while they are being heated up to the full-speed temperature. This presence of water is completely avoided if the steam is superheated, because the superheater itself is a large reservoir of heat, so that although the rate of evaporation in the boilers is very rapidly increased, yet the steam arrives at the engines perfectly dry, and in most cases superheated, thus having some surplus heat available in order to raise the temperature of the turbine itself. A further advantage of superheating is that the engine-room is much cooler and consequently more comfortable. It is always noticed during a race in which destroyers take part that those fitted with superheaters get away much quicker than those using saturated steam, and a start of a mile or two is a very valuable asset in a fight. Vessels of the *Miranda* type are about 270 feet over-all, beam 25 feet 7 inches and depth amidships 16 feet 3 inches. They have three Yarrow superheating boilers, and are fitted with two turbines of the Brown-Curtis type driving directly the propeller shafts without the intervention of gearing. Brown-Curtis turbines were applied to these vessels on account of the large clearances in this type of turbine which made the use of superheated steam quite safe. Sir Thomas Bell, Engineer-Commander Wood, and Mr. Pigott deserve great credit for the investigations they carried out at Clydebank to make these Brown-Curtis turbines a success not only on destroyers but also on much larger vessels.

In the first vessels of this class a speed of 35 knots and a radius of action of 2,500 knots at 15 knots was guaranteed. In the last vessel of the class, the *Tyrian*, a speed of 39.7 knots and a radius of action of 3,267 knots at 15 knots was achieved. This speed was obtained on the Skelmorlie measured mile in deep water, which is recognized as being the standard mile for high-speed vessels, it being well known that in water of a less depth the hull resistance is reduced at high speed and fictitious results may be obtained.

We ought to be very grateful to Messrs. Denny of Dumbarton for providing us with the Skelmorlie mile, which is acknowledged to be the best mile in the world.

High-speed vessels owe a great debt to them for providing the means of obtaining the accurate data on which future progress is based.

When speaking of the measured mile it may not be out of place to propose that a new deep-water measured mile be set up off the north end of Arran, close to the Cock of Arran, and running approximately northeast and southwest. On this course there is plenty of room at the end for the largest vessel to turn round safely and to attain the full speed before entering on the measured distance, while the depth of the water is not less than 40 fathoms.

It is sometimes overlooked how much power is absorbed in accelerating a vessel after the speed has been reduced by a turn at the end of the measured mile.

From careful surveys that have been made off the Cock of Arran, it may be possible to obtain a base 2 sea miles long, which would lead to more accurate results at high speeds. The measured distance would be in a sheltered position, clear of the traffic, and it would have the great advantage of the course being parallel to the shore. Anyone who has run high-speed vessels will know the difficulty which the pilot has in quickly getting his bearings, for when travelling at 40 knots there is not much spare time to get the correct direction after the turn at the end of the mile, and the direction of the line of the shore is a better guide than swinging compasses. The only reason for proposing a new mile is because the large vessels and high speeds of today make it undesirable to run so near the growing excursion traffic and so close to the foreshore of a residential neighborhood.

One of the most satisfactory features of war experience is that vessels of the *Miranda* class suffered little or no damage due to sea conditions. They proved quite strong enough for the various kinds of work that they were called upon to perform, sometimes under the most trying circumstances.

The *Mounsey*, under command of Lieutenant Craven, was the means of saving 696 people from H.M.S. *Otranto*. The sea was very rough at the time, and the *Mounsey* could not lie alongside the *Otranto*. She had to steam slowly past while the crew of the latter jumped to her deck, which maneuver she repeated several times. The bumping to which the destroyer was exposed during this operation can be easily imagined, but the structure was uninjured.

Another case of interest as showing the strength of these vessels was when the *Musketeer* went to rescue the crew of the *King Edward*. The *King Edward* had overhanging sponsons, and while taking the crew off in a heavy sea, the destroyer got underneath the sponsons which were descending at every roll of the ship on the deck of the destroyer and severely punishing her. Incidentally it may be mentioned that there were some high-explosive shells in the ready racks on the deck of the destroyer which caused the captain a good deal of anxiety each time the sponson struck the deck, but fortunately none of them went off. In this case several plates were actually torn away, but the main structure was uninjured, and she was quickly repaired and again ready for active service.

It is not generally realized how severe are the tests to which high-speed vessels are subjected by the gun armament. When one considers the recoil of a 4.7-inch gun on such a light structure, or the recoil of a 4-inch gun repeated 20 times a minute, it is remarkable that no ill effects are experienced. One of the tests on a gun trial is that a man should be able to keep his eye on the sighting telescope while the gun is being fired. If there is the slightest yield in the mounting, apart from the recoil chamber, it will be seen that the telescope is likely to strike the man a nasty blow. The structures are so designed that the gunlayer has no difficulty in keeping his eye fixed on the target even while the gun is being fired.

The success of these light structures is in a measure due to the uniformity of elasticity longitudinally, which is a great source of safety. Some vessels have no doubt come to grief through sudden changes in strength, and it appears that uniformity of elasticity is one of the most important points in this class of light construction.

I think you will agree with me that special credit is due to the designing staff both of ship and engine departments, which can successfully produce a high-speed destroyer, and any firm who have constructed these vessels will know the accurate workmanship which is required and obtained by special overseers appointed by the Admiralty for this type of vessel.

The direction in which further advances in speed may be looked for are as follows: (*a*) Improvements in the shape of the hull, so as to reduce the hull resistance; (*b*) improvements in the shape of the hull in wake of the propeller, so as to increase the propeller efficiency; (*c*) perfect balancing of the propeller, and the removal of the propeller as far as possible from any disturbance which may be caused by the propeller brackets; (*d*) reduction in the weight of machinery by the adoption of high-speed turbines; (*e*) reduction in the weight of machinery by increasing the output of steam from the boilers; (*f*) by the careful elimination of any items which are not necessary for the ordinary upkeep of the ship. To provide for extraordinary occurrences is handicapping the ship by carrying weight which will probably never be used; (*g*) reduction of the skin friction by eliminating all outside transverse laps, especially at the forward end of the vessel, as first pointed out by Mr. Baker.

During the war Messrs. Yarrow were also called upon to design and construct some high-speed gunboats for river purposes, known originally as the China gunboats. In such vessels the problem of speed is entirely different from that of a destroyer.

It is well known that they were ordered by Lord Fisher for use in Mesopotamia, and he gave Messrs. Yarrow an entirely free hand with respect to their design; the only stipulation being that they were to be ready for service in the shortest possible time, and that they were to be a success. Sixteen small vessels were designed and built at Scotstoun, and 12 larger vessels were designed at Scotstoun and built by Messrs. The Ailsa Shipbuilding Company, Messrs. Barclay, Curle and Co., Messrs. Lobnitz, Messrs. The Sunderland Shipbuilding Company, and Messrs. Wood, Skinner and Co. The problem was to carry a certain armament at the highest possible speed on a shallow draught, and at the same time to provide good accommodation for the crew. In the case of the smaller vessels the draught was not to exceed 2 feet 6 inches, and in the case of the larger vessels the draught was not to exceed 4 feet 6 inches. To achieve this result the vessels had to be constructed with a flat bottom, and in order to provide for propellers large enough to utilize sufficiently the power required for the speed, the vessels were fitted with tunnels, so that when at rest the propeller was only half-immersed, and when running the tunnel filled itself automatically, the propeller then working totally immersed. The after end of the tunnel when starting has necessarily to be below the surface of the water to allow for the expulsion of air and to provide an air lock, but when the tunnel is full and the water is flowing through it at a high velocity, the resistance of the vessel can be greatly reduced if the after end of the tunnel be raised up so as to be almost horizontal. This is accomplished by means of the Yarrow hinged flap. The efficiency of this flap can be best expressed by stating that for the same power 10 per cent more speed can be obtained.

The particulars of these vessels are as follows:

	Small.	Large.
Length	120 ft.	230 ft.
Beam	20 ft.	36 ft.
Speed	10 miles.	18 miles.

It is submitted that, to obtain 18 m.p.h. on a draught of 4 feet 6 inches when carrying two 6-inch guns is as good a performance as 39.7 knots in a destroyer.

When mentioning high-speed vessels there ought to be included such outstanding examples of naval architecture as the *Repulse* and *Hood*, built by Messrs. John Brown and Co., and the *Renown*, built by Messrs.

Fairfield Shipbuilding Company, which have obtained speeds never thought practicable with such gigantic ships. One cannot help feeling proud of the part which the *Clyde* has played in obtaining these various record results, and as long as there is the same cordial co-operation between employers, the scientific designing staff and the workmen, there need be no anxiety about records in the future.—"Engineering."

THE GENERAL TREND OF DIESEL MARINE MACHINERY.

Engineers who are actively engaged on the construction of any one particular type of machinery, are liable to become so engrossed in their own limited sphere of activities and in meeting the problems and questions with which they are immediately confronted, that they do not make the opportunities, or spare the time, to take cognizance of and to review periodically the broad trend of affairs in the same line of work the world over. It is, however, necessary in order to reach or to maintain a high position in any branch of industry, that full recognition should be given to what is being accomplished elsewhere, and to learn as much as possible from successes or failures of similar products in markets other than those catered for by home engineers.

There is, at the present time, no problem concerning the minds of our marine engineers to a greater extent than that of the application of the Diesel oil engine to marine propulsion. The development of this prime mover is international, and whilst economic conditions principally, together with a few other subsidiary factors, have contributed hitherto to a larger and broader activity of construction for both land and marine purposes abroad than at home, events are now rapidly marching to the desirable position where the technical and commercial competition between the various maritime and shipbuilding countries will be extremely keen.

It is, therefore, now opportune to review briefly and on general lines the characteristics of the types of engine which are on order and under construction the world over. Statistics, so far as they are available and without any data as to German building, show that in the field of ships of greater deadweight capacity than 2,000 tons, considerably more than 150 are on order, and in course of construction at the present time, and are destined to be fitted with marine Diesel engines. In some cases the ship is being built in one country whilst the engines are to be supplied from another, but since relatively few novelties in the shipbuilding sense are presented by these vessels, this review will be entirely concerned with considerations of the propelling machinery. The Diesel motors for these ships are being supplied by at least seven countries, and in respect of numbers of sets of machinery the order of importance is approximately the following: Denmark, United States of America, Britain, Sweden, Holland, France, Italy and Switzerland. Since the numbers upon which this sequence is based include definite future as well as present construction, those countries which early on took a leading part in the marine Diesel world, can naturally obtain and can safely book orders much further ahead than can be or would be desirable in the case of the countries where a later start was made or the interference of the war had effect. With this proviso, however, the statement of the order is significant, and probably indicates quite approximately the extent of development of the Diesel marine engine in the various countries named.

The position of Britain cannot be regarded with complacency. In justification, if not in full measure, we can plead the effects of the war

which seriously suspended construction other than of a thoroughly standardized nature in order rapidly to give results with the minimum of expenditure of workmanship and materials. In such circumstances the steam engine naturally was the prime mover upon which to concentrate. Undoubtedly, however, at the present time our position is rapidly being improved, and there are other factors later to be touched upon which can also be quoted in mitigation. It will be noted that of the first five countries in the above list three were neutrals in the war, and in the case of the United States of America, the effects of the war were only partially felt and at a much later date than in the United Kingdom.

A review of the type of engine being constructed goes to show that the simple single-acting, four-stroke cycle engine is even more favored today in connection with marine Diesel work than it has ever been in the past history of this type of mercantile construction, and of the vessels building only some 16 per cent are to be fitted with engines of the two-stroke cycle type.

It will readily be conceded that this figure is quite remarkable. If numbers of engines were taken, the percentage of two-stroke cycle construction would probably even be considerably lower, since almost half of the two-cycle engined ships are single-screw, whereas 90 per cent of the four-stroke cycle machinery is of the twin-screw variety. In the early days of the introduction of the oil engine to marine work, the two-stroke cycle was generally the favored system. The same success as with the four-stroke cycle engine did not meet these initial efforts, and these failures are, in a measure, no doubt reflected in the present strong position of the four-cycle engine. In power per cylinder or per engine the average today with the two-cycle is higher than with the four-cycle engine. The reason for the preponderance of twin-screw ships is the desire to attain to total powers installed aboard comparable with the average steamship, and to achieve in propeller efficiency, a figure approaching that of slow-running steam engines. Of all the motor vessels over 80 per cent are fitted with twin-screws.

It is interesting, further, to reflect that of the relatively few single-screw vessels in construction, half are propelled by two-stroke cycle engines. The reasons for this may be firstly, that some of the largest powered units are of the two-stroke cycle type, and constructors who are willing to take the risks incumbent upon fitting the newer, less tried-out and less conservative type of engine would seem also to be willing now to take whatever risk, if any, there is in the policy of one marine Diesel engine per ship. The two-stroke cycle engines comprise the opposed-piston types at present exclusively built in Britain, the supercharging port scavenging engine, originated in Switzerland and being built now chiefly in Switzerland and Italy, and the valve-scavenging engine which is only now retained by one French constructor.

In regard to the characteristics of the various countries, it is interesting to note that in Britain by far the greatest variety of engines are being constructed, these comprising standard four-cycle air-injection and four-cycle solid injection engines, two types of opposed-piston Diesels and supercharging two-cycle engines, and also single simple port-scavenging two-cycle engines. Sweden, Denmark and Holland are solidly in favor of the four-stroke cycle engine, and in America, very few two-cycle Diesel engines are in course of construction. In Switzerland and Italy are to be found the strongest protagonists of the two-cycle type.

In Britain not only is the greatest variety of type evidenced, but also the largest engines are being constructed, the most powerful, the four-cylinder opposed-piston two-stroke engine of 750 brake horsepower per

cylinder. The cylinder diameters are 580 mm. and the stroke 1,160 mm., and the engine develops, at 70 revolutions in the four cylinders, a total of 3,000 brake horsepower. The largest four-cycle engine being constructed in Britain, and also elsewhere, has cylinder diameters of 740 mm. by a stroke of 1,150 mm., and runs at 115 r.p.m., developing over 3,500 brake horsepower in eight cylinders. The two-cycle engines are of four and six cylinders—generally of four cylinders—and the four-cycle engines have six cylinders with a few exceptions where eight-cylinders are employed to give the desired high power.

The British policy of trying out all types of engine is certain to lead in the end to a very favorable position, and is a very bold policy to be commended in view of the lee-way which had to be made up and the lead unquestionably held by the earlier constructors of four cycle engines. The unfortunate and unlikely event of failure with any of the newer types cannot now, in view of the varied and long experience already piled up to the good, be taken as evidence sufficient to condemn this prime mover in respect of its suitability for marine work.—"Engineering."

EXAMINATION OF LUBRICATING OILS.

If the multiple and complex problems of oil testing are to be entrusted to the engineering chemist, as would appear to be reasonable on the whole, then the expert so acting must be proficient in both his professional branches and possess a good knowledge of physics in addition. One expects a great deal of the oil chemist. It is his duty to advise us how to extract the oil or how to produce it by other means, to tell us what the oil consists of, and how it should be treated to yield constituents originally in the oil or to be formed in the course of the treatment to which it is submitted, and how to modify the method of treatment in accordance with the varying demands for special products. He should also understand the preservation and storage of oils. The routine testing of oil is only a part of his duties, which he can, and must, to a certain extent, delegate to assistants. As a consequence, varieties of tests have been worked out and have come into practical use which can hardly yield concordant results when performed by different workers, and are at the best of doubtful value. Hence the growing demand for standard tests. But there is little agreement, so far, as to the co-ordination of the results, and even standardized tests would only supply the inquirer with certain figures or constants without telling him whether or not oil will answer his particular purpose.

That the Institution of Petroleum Technologists is anxious, in its own interests, to assist in the elucidation of the problems was emphasized by Sir Frederick W. Black, when taking the chair at the meeting on Tuesday, April 20, the second meeting during this session, which the institution devoted to oil examination. In October last Mr. Arnold Phillips had discussed "Laboratory Tests of Mineral Oils;"* on April 20, Mr. G. F. Robertshaw, F. C. S., of Manchester, presented a paper on "Methods of Examinations of Lubricating Oils." Standardization and lubrication are very much to the front at present, and will remain so—on experimental lines in the first instance, we hope. At the October meeting Sir Thomas Holland had expressed the opinion that the experimental work would

* See Engineering, October 31, 1919, page 586, and the discussion on "Lubrication" at the Physical Society, Engineering, December 5, 1919, pages 755 and 758.

have to be paid for. Mr. Robertshaw drew attention to the way in which the Society of Leather Trades Chemists in England and the Leather Chemists Association in America deal with their own, likewise complex, standardization problems. Sectional committees are elected to take up particular lines of research, mostly without expense to the society. To a large extent that has been done also under the general guidance of the Lubrication Committee without restricting the investigator too much as to following up novel lines of research, whilst in the case of paid investigations certain limits have in general to be imposed.

In reviewing the properties and examination of lubricants Mr. Robertshaw distinguished between oils, greases, and special lubricants, classing the oils again as fixed oils and mineral oils purely of the hydrocarbon type. The collective term "fixed oil" is also used to distinguish the non-volatile oils produced by expression or extraction from the volatile essential oils produced by steam distillation. Fats, tallow and lard belong to the fixed oils and are complex organic compounds, either glycerides (*i. e.*, compounds of the salt type of the fatty acids with the alcoholic radicle glycerol or glycerin), or esters (compounds of the oxide type). The glycerides are saponified by the action of alkalis, *i. e.*, split up into glycerin and soaps, salts of the fatty acids; they are also decomposed or hydrolysed by other agents. Free fatty acids may form soap with the metal with which they come in contact, but Wells and Southcombe* have not observed any metal corrosion by the free fatty acids which they add to mineral oils in order to raise the lubricating power. Though few of the fixed oils serve directly and without admixtures as lubricants, some of these enjoy high favor; but castor oil (ricinus) is now being replaced by mineral oils even on aeroplane engines. Rape seed oil shares with other vegetable and animal oils a certain inconstancy in composition and viscosity; yet it still serves as standard liquid for the Redwood viscosimeter. Blown rape oil and other oils oxidized by hot air are used for compounding oils. Greases are partly mixtures of soaps (sodium and calcium) and of refined mineral oil products. The special lubricants comprise pastes wanted for screw cutting, &c., for textiles, leather, &c., and graphite, talc and steatite, &c., used alone or mixed with mineral oils to which they are to impart a fictitious viscosity. Among the new synthetic lubricants developed during the war in Germany Mr. Robertshaw mentioned anthracene oil, which is prepared in thousands of tons.

Many of the properties and tests of these lubricants, important for identification and evaluation, have only an indirect interest for the engineer. Color matters for the textile manufacturer; density and boiling point are of no importance as such, within certain limits. Surface tension might prove to be all-important, if we could only measure the surface tension between the oil and the metal; what the various methods of determination now in use really signify, we do not understand. The cold test tells us whether the lubricant is apt to thicken or to form a deposit at low temperature; it may be performed by gradual cooling (and rewarming) the stirred oil and by watching the formation (or disappearance) of deposits, or, in the case of turbid oils, in the pour test, by noting when the flow of the oil ceases or recommences. The acidity tests and iodine value indicate liability of the oil to chemical reaction.

The tests of volatility, flash point and gumming are likewise strongly influenced by chemical reactions, and hence depend very much upon the actual experimental conditions. The volatility test (important for internal combustion engines) determines the loss by evaporation at definite

* See Engineering, February 6 last, page 186 *ante*.

temperatures within certain time intervals. Quoting Redcliffe, Mr. Robertshaw did not consider any of the many appliances in use as reliable, though in cases of small loss—and oils of high volatility are hardly lubricants—the Archbutt-Deeley apparatus gave fairly concordant results. Dr. Ormandy added in the discussion the interesting information that German chemists had recently observed greater heat losses when the test was conducted in a nitrogen atmosphere than when performed (as usual) in air or steam, and that he had himself found on continued heating that, whilst the flash point rose as the heat loss decreased, the curve oscillated very strongly. Neither observation is surprising, but they both exemplify the difficulties of the tests. The gumming test (performed with nitro-sulphuric acid, petroleum ether, alcoholic soda, or simply by heating a shallow layer of the oil in air) is clearly one of the tests which cannot generally be standardized for all kinds of oil, and is, perhaps, not much in favor. Dr. Thole mentioned, however, that he had by this test (simple oxidation in air) discriminated between a good and a bad motor-cylinder oil; identical practically as to evaporation, flash point and viscosity, the one of these oils gave 0.4 per cent, and the other 12 per cent of residue. Both the emulsification tests (necessary in the case of steam turbines particularly) to which Mr. Robertshaw referred, those of Mr. A. Philip and of the Bureau of Standards, have been described in our columns.

There remains the most reliable physical test of lubricants, the viscosity test.* Describing recent modifications of tests in use, including the Michell viscosity meter (see "Engineering" of April 16 last, p. 509), Mr. Robertshaw recommended the adoption of the poise (the name is taken from Poiseuille) or the centipoise (0.01 poise) as a practical unit of viscosity in absolute measure. The centipoise approximates very nearly to the viscosity of water which is the comparison liquid both of the Engler and the Saybolt viscosimeters, the standard instruments in Germany and the United States. Some unification in this respect is very desirable, and we have stated already that the rape-seed oils, the comparison liquid of the Redwood standard viscosimeter, as now obtainable fluctuates strongly in viscosity.

As regards the relation between viscosity and lubricating value Mr. A. Philip remarked that once a certain minimum value of viscosity, necessary to prevent squeezing out of the oil, is exceeded, the high viscosity goes together with low lubricating value; "oiliness" he would simply define as lubricating power, to be measured by friction tests in want of any adequate physical or chemical tests. Mr. Robertshaw had expressed a similar opinion when describing the Thurston machine and showing some blue prints of a large mechanical tester built by Sir W. Bailey and Co. for the Government, and designed for bearing speeds of 10,000 feet per minute at pressures up to 120 pounds per square inch as well as low speeds of 6,000 feet at 500 pounds per square inch. Dr. A. E. Dunstan suggested that oiliness like other abnormalities could be traced to unsaturation and to colloidal structure. To Dr. Ormandy the colloidal character of lubricants suggested a possible experimental method of determining the bonding between the metal and the lubricant. When clay (a colloid) was mixed with a small proportion of caustic potash, the clay set and the water filtering off was free of soda; when, however, a potential of 1.6 volt was applied to the paste, soda appeared in the filtrate: thus we have an electric measure of the absorption of the alkali by the clay. Mr. E. H. Cunningham Craig said that he did not see any difficulty in

* See Engineering, April 19, 1918, page 433 (Dunston and Thole), June 17, 1918, page 655, and September 18, 1918, page 294 (Higgins and Herschel).

connecting oiliness with the varying surface energies, probably electric, of the various constituents of the very complex lubricants. When a surface was brought against an oil of that kind, there was a concentration along the solid surface of the particular disperse phase which has a surface energy nearest to that of the solid, followed by the selective concentration of other disperse phases one above the other, until a perfect arrangement preventing internal friction was reached; that was oiliness. Further information on Deely's oiliness test was not given during the discussion, and most people probably advocate standardization of oil-testing appliances simply to get over the great confusion prevailing, rather than because they have a decided belief in the efficacy of any of them. Our conclusion hence remains that any standardization, if it is to be really useful, will have to await the outcome of further experimental research.—"Engineering."

ESTIMATES OF PROPELLER PERFORMANCE FROM SELF-PROPELLED MODEL TESTS.

By Commander William McEntee, Construction Corps, U. S. N.

During the last four years there has been developed at the Experimental Model Basin at the Washington Navy Yard an interesting and valuable method of testing the models of ships with the propellers in place and actually propelling the model. Sufficient experience has been had to indicate that the method is capable of giving very accurate results. It also has been of much value to the Bureau of Steam Engineering in supplying an independent check on propellers designed by Dyson's well-known method. The subject should, therefore, be of interest to naval architects and marine engineers who in the estimate of power requirements for new ships must take two important steps in the design of a ship and its propelling machinery:

(1) The selection of suitable lines for minimum resistance and the estimate of the effective horsepower required at different speeds.

(2) The selection of a suitable propeller so that the power of the machinery may be applied to the propulsion of the ship most effectively.

When there is a happy solution of both requirements (Nos. 1 and 2), the resulting design is one which is economical in both fuel used and in weight of the machinery installation.

As regards the determination of the form of the hull, experience in the use of Froude's law of comparison in testing models in model tanks in different countries has amply demonstrated its accuracy. It is in fact the only satisfactory method of estimating the effective horsepower required for a ship when its lines differ in a material degree from those of previous models and ships tested. Moreover, there has been a large amount of experimental data made public with regard to the factors affecting a ship's resistance, so that with intelligent application of the same a fairly close estimate may be made for ships of normal type.

When it comes to the design of a propeller to use on a particular ship, there is inherently a more difficult problem to solve. The performance of a propeller alone, even when running in free undisturbed water, is dependent on so many variables that it is a very complex subject. When to these complexities of the propeller alone are added those due to the form of the ship's hull itself, it is readily seen that there is greater need for an accurate model-testing method for estimating power requirements for a given combination of hull and propeller than there is for estimating the performance of either by itself.

For the propeller alone, the experimental field has been well covered in the researches of Froude in England and Taylor and Durand in this country, as well as by those of other investigators in Germany and elsewhere. These experiments have served to clear up to a large extent the haze which until comparatively recently seemed to surround the conceptions of propeller action. In fact, the experimental data derived by these investigations have served to discredit many mathematical theories of the propeller, so that at the present day, so far as the writer is aware, there is not a single rigidly rational propeller theory which fully explains the observed phenomena or which may be used with certainty to predict the performance of a new type of propeller.

It may be considered, then, that the information available as to the lines of ships' hulls and the performance of propellers of conventional form in free water are sufficient for all ordinary purposes. It is when seeking to estimate the effect of combining a certain propeller with a certain hull that there are introduced elements of doubt which may lead to important errors in estimates of shaft horsepower or revolutions of the propeller.

Recognizing this difficulty, Rear Admiral Dyson has developed a method of design which is based on the trial results of actual ships. The estimates obtained by his method are entirely satisfactory when the ship and propeller fall in the range of variables covered by existing trial data. It is apparent that, with the almost unlimited number of possible variables in form of hulls and propellers, it will require many years and the trials of many ships before it becomes of universal applicability.

There is, therefore, a real necessity for a method of testing models of the hull and propeller running together. Experimental work of this kind has been done by Froude and Luke in England, but their published results are limited to statements of certain factors, such as wake and thrust deduction coefficients resulting from certain combinations of hull and propeller. There are no data available to show how close the agreement between the performance of the full-size ship and those estimated from the model tests is. The arrangement of the test apparatus used by them is open to the criticism that it does not completely fulfill the requirement of similarity between ship and model, as the propeller is not carried by a shaft extending from inside the model, but by a separate shaft wholly outside of it and supported by separate attachment to the towing carriage.

The following is a description of the method as developed and used at the Washington Model Basin with comparative results of model tests and ship trials for the U. S. S. *New Mexico*.

The model to one-thirtieth scale was fitted with all appendages and with shafts, struts and propellers to scale in accordance with plans of the ship and machinery as built. The appearance of the model fitted with propellers is shown in Figs. 1 and 2.

In making the tests each shaft was driven by a small direct-current motor coupled directly to the shaft. Each motor was arranged as a dynamometer with calibrated springs which permitted simultaneous measurement of torque, revolutions per minute and thrust on each shaft, while the model was self-propelled under the Model Basin towing carriage. The model was arranged so that the towing carriage would steer in a straight course while running at a uniform predetermined speed. The model was free to gain or lose a distance of about six inches as compared with the towing carriage before it would strike stops. During the course of each run the speed of the propellers was adjusted so that the model was running under its own power at a speed just equal to that of the towing carriage. When the model was running uniformly, so as not to strike either stop, cards recording the revolutions per minute, thrust

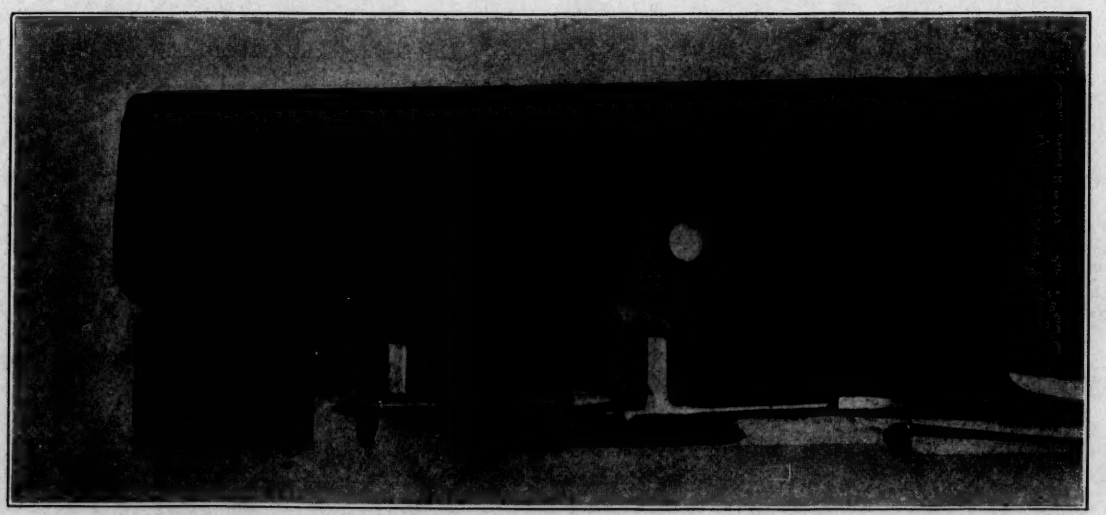


FIG. 1.—SIDE VIEW OF STERN, SHOWING ARRANGEMENT OF SHAFTS, STRUTS AND PROPELLERS AS TESTED IN THE MODEL BASIN.

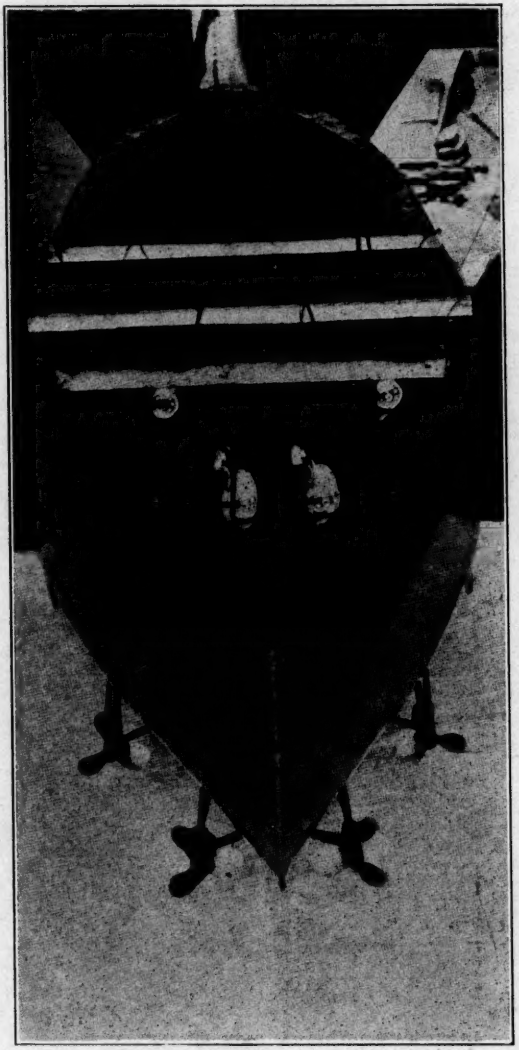


FIG. 2.—STERN VIEW, SHOWING SHAFTS, STRUTS, PROPELLERS AND DYNAMOMETERS IN MODEL AS TESTED IN THE MODEL BASIN.

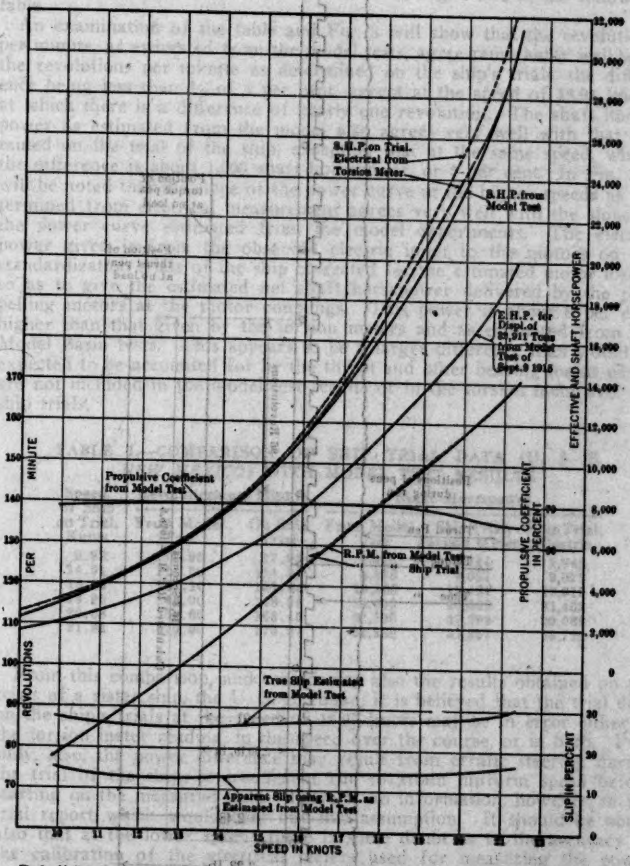

S.H.P. on Trial. Electrical from Torsion Meter
S.H.P. from Model Test
E.H.P. for Displ. of 28,911 Tons from Model Test of Sept. 9 1918
Propulsive Coefficient from Model Test
R.P.M. from Model Test
" " Ship Trial
True Slip Estimated from Model Test
Apparent Slip using R.P.M. as Estimated from Model Test
REVOLUTIONS PER MINUTE
EFFECTIVE AND SHAFT HORSEPOWER
PROPULSIVE COEFFICIENT IN PERCENT
SLIP IN PERCENT
SPEED IN KNOTS

Fig 3.—U. S. S. *New Mexico*. Comparison of Power and Speed Data on Standardization Trials with Results of Self-Propelled Model Tests (Model No. 1642). Standardization Trials of December 16 to 18, 1918. Length on Load Waterline, 600 Feet; Beam, 97.156 Feet; Draft, mean, 30.825 Feet; Displacement, 32,911 Tons; Diameter of Propeller, 13 Feet 5 Inches; Pitch of Propeller, 15 Feet 2 Inches; Ratio Projected Area to Disk Area, 0.3562; Number of Blades, 3; Number of Propellers, 4; Scale of Model, 1/50 Full Size.

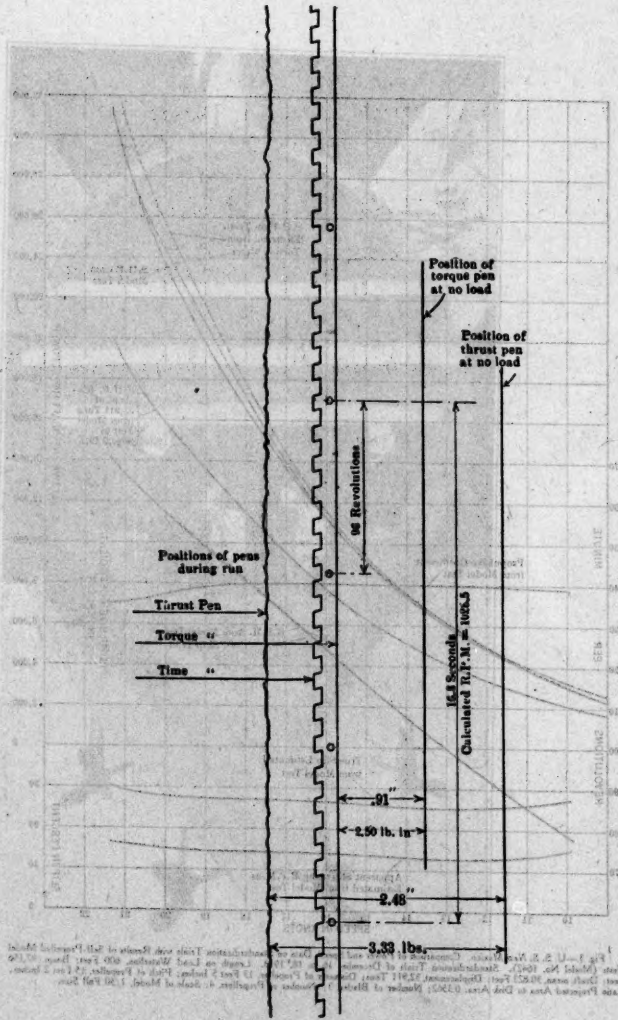


Fig. 4.—Dynamometer Card, Showing Propeller Thrust. Torque and Revolutions per Minute

and torque on each shaft were taken simultaneously. The speed of the model was taken by a card on the towing carriage in the same manner as when making model resistance experiments. A sample dynamometer card is shown in Fig. 4.

The results of the experiment are shown in Fig. 3 and in the following table.

An examination of the table and Fig. 3 will show that the revolutions per minute, as estimated from the model tests, agree remarkably well with the revolutions per minute as determined on the ship's trials, the difference being less than ½ of 1 per cent, except at the speed of 18.95 knots, at which there is a difference of nearly one revolution. The shaft horsepower as estimated from the model also agrees very well with that obtained on the trial of the ship, excepting that at the same speed, where the difference is about 1,000 shaft horsepower, or 5 per cent. In Fig. 3 it will be noted that the slope of the power curve at the higher speeds as determined from electrical measurement agrees very well with the slope of the power curve estimated from the model experiments. The electric power given is from the observed electric input to the motors on the standardization trials of the ship corrected for the estimated motor losses, so as to give the estimated net shaft horsepower delivered by the propelling motors at the motor couplings. This power is about 8 per cent higher than that given by the torsion meters and as estimated from the Model Basin tests. This appears to be a larger difference than would be expected to be accounted for by the thrust and other bearing losses which are not included in the model test results or in the torsion meters on the ship trials.

TABLE I.—COMPARISON OF SHIP TRIAL DATA (U. S. S. *NEW MEXICO*) WITH MODEL TEST RESULTS

Speed of Ship on Trial, Knots	Revolutions per Minute		Shaft Horsepower		
	From Model Test	On Ship Trial	From Model Test	Ship Trial, Torsion Meter	Ship Trial, Electric
9.92	77.95	77.92	1,815	2,144	2,948
14.91	114.50	114.89	5,818	9,021	9,072
16.84	131.19	131.35	12,285	13,481	13,919
18.95	148.00	148.94	19,600	20,662	21,408
20.68	163.60	163.45	27,190	27,292	29,088
21.81	170.00	170.27	31,290	31,357	33,722

From this comparison, and considering also the results obtained on the trials of a sister ship, the U. S. S. *Idaho*, it is believed that the trial data on the ship's trials at the speed of 18.95 knots may be in error either in the torsion meter reading, in the speed over the course, or in both. Possibly, also, the power difference may result from erratic steering during the trial of the ship, or insufficient run to attain uniform speed before starting on the measured mile. There is no information, however, in the trial report which would bear out this assumption. It should be noted also that at the lower speeds there is some doubt as to the accuracy of the calibration of the electrical meters used for measuring the power input to the motor.

In estimating the shaft horsepower for the ship, the effective horsepower for the model with full appendages, as determined in the Model Basin tests dated September 9, 1915, was used. This effective horsepower curve does not take into consideration the resistance of the paravane gear which was rigged on the ship during the trials. From the fact that the shaft horsepower estimated from the Model Basin tests and the revolutions per minute are in close agreement with the results obtained on the

trials of the ship, it is apparent that either the paravane gear did not add much to the ship's resistance, or that the effective horsepower originally estimated for the ship was high by an amount equivalent to the resistance of the paravane gear.

The average wake factor for the four propellers at a ship speed of 21 knots was found to be .15, with 16 per cent apparent slip and 29 per cent true slip. With the total power on the model distributed equally between the four shafts, it was found that the outboard shafts ran at about 1 per cent higher revolutions per minute than the inboard shafts. The propulsive coefficient is 65.3 per cent at 21 knots with a hull efficiency of 100 per cent, indicating that the arrangement of shafting, struts and hull clearances for the propellers are very good.

Another conclusion from the investigation and from other similar tests is that the Model Basin method of testing with model propellers fitted to drive a model of a ship may be used with confidence to predict the performance of the ship on trials. There is a limitation, however, in the case of propellers which on the ship are running in the cavitating region, in which event cavitation may be experienced on the ship and not occur on the model.

In addition to the work of this character already done on models of naval vessels, several progressive private shipbuilding interests of the country have been quick to see the advantages of self-propulsion model tests and have arranged to have such tests made on the models of proposed new merchant ships. The law under which the Model Basin was established provides that private shipbuilding companies may have models tested on payment of the actual cost of the work to the Government.

The present cost for making a 20-foot model and testing it for resistance at three or four different conditions of draft is $500 (104/3/4). From these tests the corresponding effective horsepower curves for the ship are computed and given in the report of test. The additional charge for fitting shafts and propellers to the model and making self-propulsion tests in the case of a twin-screw ship is about $300 (62/10/0).

From the latter tests the shaft horsepower, revolutions per minute and propulsive coefficient for the ship at different speeds are estimated and given in the report.—"International Marine Engineering."

CURRENT TENDENCIES IN NAVAL DESIGN.

Of the many warships now under construction in the United States, the six battleships of the *Massachusetts* class are, in some respects, the most remarkable. Concerning the new battle-cruisers, we may say, without risk of offending American susceptibilities, that they are slightly enlarged copies of H.M.S. *Hood*. Their bulge protection, however, is complete, instead of only partial, as in the British prototype; they are to steam at 33¼ knots, as against the *Hood's* 31 knots; their eight 16-inch guns fire a projectile of 2,100 pounds, compared with the 1,920-pound projectile of the *Hood's* 15-inch weapons; and their secondary armament of sixteen 6-inch quick-firing guns is superior in numbers and weight of fire. But, generally speaking, they present no novel features. It is otherwise with the battleships, which are distinctively American in conception. The comparison, given elsewhere, of their design with that of the *Hood* brings out very clearly the great forfeiture of offensive and defensive power which high speeds involve. It will, no doubt, be urged that comparisons between warships of fundamentally different type are misleading, that for

the tactical functions which the *Hood* is designed to perform, high speed is a *sine qua non*. On the other hand, it is open to question whether the value of speed—tactically, if not strategically—has not been exaggerated. War experience has done much to modify naval views on this subject, as may be gathered from various passages in Lord Jellicoe's book, from the remarks made by Captain Sir Edward Chatfield at the Institution of Naval Architects last month, and from the writings and utterances of other distinguished officers. Captain Chatfield's words are particularly significant, for this officer, who served as flag-captain to Admiral Beatty throughout the war, and took part in every major engagement with the exception of the Falklands battle, is exceptionally well versed in modern tactics. Discussing the qualities of the *Hood*, he declared that those who had fought at sea had not found speed to be the most important feature in a warship. The main idea underlying superior speed was that it enabled the captain of the ship possessing it to choose the fighting range, but "that assertion is far from being true." Those who had been engaged in naval warfare knew, he continued, that the fighting range was largely determined by the weather. If the range of visibility was only 10,000 yards, it was useless to possess the speed and gun-power to enable the action to be fought at a range of 25,000 yards. "That case had happened over and over again." There could be no doubt, he added, that if new warships were to be designed today, British naval officers, at least, would not design another *Hood*.

If we rule out the British 18-inch gun, which is no longer mounted in any ship, the 16-inch 50-caliber weapon with which the *Massachusetts* and her sisters are to be armed must be considered the most powerful naval gun of the present day. Except that the projectile weighs 2,100 pounds, and the muzzle energy developed is about 115,000 foot-tons, nothing is known of the ballistics of this piece. Provided, however, that the mountings permit of the requisite elevation, it should certainly be able to outrange both the 14-inch and 15-inch pieces carried by contemporary battle-cruisers, and in that case the superior speed of the latter would be useless except as a means of escape from the crushing salvos of the slower ship. And when all is said, capital ships, be they battleships or battle-cruisers, are primarily built to fight, not to fly. There are indications that the naval world is readjusting its ideas as to the relative values of armament, armor, and speed. We have lately heard more than one authority recant his pre-war belief in the supreme value of speed. Naval opinion, both here and in the United States, appears to be crystallizing in favor of an ideal type of capital ship comparatively moderate in speed, and therefore, *mutatis mutandis*, in displacement, but embodying the maximum degree of hitting and resisting power possible within the limits of that displacement. That we shall ever revert to the displacements of the pre-Dreadnought era is practically out of the question, for big guns and thick armor necessitate generous dimensions, quite apart from the increase due to high speed. But it is possible that the capital ships designed a few years hence will be considerably smaller than either the *Massachusetts* or the *Hood*. If it be true that the leading authorities on both sides of the Atlantic have come to the conclusion that high speed is not a prime tactical necessity, and that its possession in no way compensates for inferiority in armament or protection, the way is already open for a large decrease in the size of capital ships. At the same time, we are not over confident that the opportunity will be seized. The constructor, encouraged by the naval officer, will assuredly be tempted to turn the weight saved by reduced engine power to account in some other direction rather than content himself with a smaller ship. He will go on adding to the number and caliber of the guns, or to both, and to the thickness of the

armor protection, until the margin of surplus weight is quite swallowed up and a ship larger than ever results. And when that stage is reached it is possible that high speeds may again come into favor, in which case a further huge jump in dimensions will become inevitable.

We would like to add a word of protest against the practice of comparing the resisting qualities of pre-war British and German capital ships to the exclusion of other features. It is true that the German ships were better able to withstand punishment, above and below water, than our own, but the cause of this discrepancy was strategical rather than technical. Germany, as has since been admitted, planned her ships mainly, if not solely, for the purpose of fighting a North Sea campaign. They were never designed to make long cruises or to accept battle in waters remote from the German Bight. They were, in effect, very large coastal ironclads, in designing which it was not necessary to waste much space on living accommodation, since their officers and men were intended to spend most of their time ashore, and an unusually large percentage of displacement and area could therefore be devoted to armament and protection. Our ships, on the contrary, were designed in accordance with the Navy's immemorial tradition of ability to cruise and fight in any quarter of the navigable globe. Consequently the factors of habitability, hygiene, and comfort could not be ignored, as, broadly speaking, they had been in the case of German ships.—"The Engineer."

SHIPBUILDING IN JAPAN.

The new Japanese battleship *Mutsu*, which was laid down in 1918 at the Yokosuka Imperial dockyard, was successfully launched on the 31st of May. This vessel is described as the largest in the Japanese Navy, from which it would appear that her sister ship, the *Nagato*, being built in dry dock at the Kure dockyard since August, 1917, has not yet been floated out. The type they represent is undoubtedly a most formidable one, and to those who are out of touch with current naval progress it may come as a surprise to learn that the *Mutsu* and *Nagato* are more powerfully armed than any ship in the British Navy, and larger than any warship now afloat, with the sole exception of H.M.S. *Hood*. Their chief dimensions are:—Length, 661 feet; beam, 95 feet; draught, 30 feet; displacement, 33,800 tons. Geared turbines are to be installed in both ships, from which a speed of 23½ knots is anticipated. The main armament will consist of eight 16-inch guns. The Japanese ships, therefore, are slightly larger and considerably faster than their American contemporaries of the *Maryland* class, the nameship of which was launched on March 20th; but the main armament is the same in caliber and number of guns. Besides the *Nagato* and *Mutsu*, six further capital ships are under construction or about to be laid down in Japan, a number that will bring the establishment up to seventeen Dreadnoughts or, in popular parlance, "super-Dreadnoughts," for all save one of the seventeen are armed with guns exceeding 12 inches in caliber.

The exact position of the Japanese naval programme at the present moment is somewhat obscure. According to a Government measure introduced late in 1917, a sum of £30,054,800, spread over a period of six years, was to be spent on the construction of two battle-cruisers, three light cruisers, twenty-seven destroyers, and forty-eight submarines. Of this amount, £2,544,000 were to be disbursed in 1918, £4,771,766 in 1919,

£5,412,381 in the current year, £6,819,635 in 1921, £7,122,673 in 1922, and the balance in 1923. Apparently, however, the programme was subsequently modified to include a larger proportion of capital ships. The ultimate aim of the Japanese Government, as explained at the time, is to realize the so-called "eight-eight" system, that is to say, squadrons each consisting of eight battleships and eight battle-cruisers, and to create in time at least three squadrons of this composition. But, irrespective of financial considerations, there are factors which augur none too well for the attainment of this goal within the measurable future. Since the close of the war shipbuilding in Japan has been seriously hampered by the difficulty of obtaining steel. She is dependent primarily on the United States and Great Britain for her supplies of this material, and within the past two years her imports from both countries have fallen off considerably, owing, on the one hand, to transport difficulties in the States, and, on the other, to the fact that the unfavorable exchange situation makes it more profitable for British manufacturers to dispose of their steel in the European markets. So serious has the shortage become that this year's output of tonnage in Japan is expected to be less than 600,000 tons instead of the 800,000 tons forecast in January. This scarcity of steel is probably reacting on the naval programme, and may account in part for the delay in floating out the *Nagato*. When the war boom in shipbuilding was at its crest new yards sprang up in Japan with mushroom-like rapidity. Just before the Armistice no less than seventy shipyards were at work in the Osaka district alone, a number that has lately fallen to twenty. The home steel supply was gravely prejudiced at the beginning of the year by an act of sabotage perpetrated at the Yawata works, the only concern in Japan which is capable of producing large quantities of steel. So extensive was the damage that the works had to be closed down for several months. Another factor tending to retard the output of tonnage, naval as well as mercantile, is the higher rate of wages demanded by the shipyard workers. Four years ago the average daily wage of a Japanese shipwright was about a yen (2s.). The present-day average is doubtful, but it is certainly much higher, and fresh claims have lately been put forward on behalf of the workers. Thus the immense advantage which cheap labor gave to Japanese shipbuilders over their rivals in the Occident is gradually disappearing.

So far as technical efficiency is concerned, Japan unquestionably stands in the front rank as a shipbuilding nation. During the period of intensive construction brought about by the depredations of German submarines, many new "records" for quick building were made in the United States, and a yard at Ecorse, Detroit River, claimed to have "licked creation" by delivering the S. S. *Crawl Keys*, of 2,300 tons gross, twenty-nine days after the keel had been laid. This achievement, creditable as it was, has, however, been surpassed by the Kawasaki dockyard at Kobe in the case of the steamer *Raifuku Maru*, of 5,800 tons gross. The keel of that vessel was laid on October 7th, 1918; she was launched on October 30th, and her official trials were successfully completed on November 5th. In her case, therefore, the building period also covered twenty-nine days, but she was larger than the American vessel by 3,500 tons, a difference that makes the record much more noteworthy. In speed of naval construction the Japanese yards are equally well to the fore. The 31,300-ton battleship *Ise*, built by the Kawasaki Company, was completed in thirty-one months. Twelve 700-ton torpedo-boat destroyers, ordered in Japan by the French Government at the beginning of 1917, were all in service by the following August, some of them having been built so quickly that the average period for the twelve worked out at five months. In the light of these

performances the assertion of a leading Japanese yard that, provided the necessary material was forthcoming, they would be prepared to deliver a battleship of the largest dimensions within twenty months from the laying of the keel, a light cruiser within eleven months, and a large destroyer within five months, does not appear to be exaggerated.

The rise and development of modern shipbuilding in Japan furnishes one of the most striking instances of the national adaptability. The squadrons which carried the Japanese flag to victory at the Yalu in 1894 and at Tsushima in 1905 had been constructed almost entirely in foreign shipyards. It was not until 1905 that the first large armored warship was laid down in a Japanese yard. But since that date more than a score of fine battleships and cruisers have been built and completely equipped by the national industry, and at the present moment, as we have seen, battleships of unprecedented size and power are approaching completion in Japan. The growth of the late Imperial German Navy during the twenty years preceding the war was regarded as phenomenal; but considering the great difference in wealth and productive facilities of Germany and Japan respectively, the latter's achievement in the same sphere of endeavor is even more astounding.—"The Engineer."

OUR NAVAL STANDING.

The fact that we are the only nation which has continued since the war to enlarge its battleship strength made it inevitable that the question of supremacy of the sea would arise and form the subject of discussion—in other words, that our naval policy should stimulate instead of helping to repress that race in naval armaments to which it was hoped that the extinction of the German navy had put an end. Evidence of this thought and talk is found in an article by Archibald Hurd in a recent issue of the "Illustrated London News," in which a very striking comparison is made of the strength in capital ships of our own navy and that of the British, when our present program of construction shall have been completed, that is to say, about the year 1924.

Except for the finishing up of a few of the smaller types of vessels, Great Britain has no construction on hand and has no thought of laying down a new program. The same is true of France and Italy. In this country, at the time the article was written, we had eighteen capital ships authorized or under construction, each of which will be larger and more heavily armed than any existing ships in the world. The battleships include the *California* and *Tennessee*, of 32,300 tons, mounting eight 16-inch and fourteen 5-inch guns; the four battleships of the *West Virginia* type, of 32,600 tons, mounting eight 16-inch and fourteen 5-inch guns; and lastly, the six battleships of the *Indiana* class, each of 42,300 tons displacement, mounting twelve 16-inch and sixteen 6-inch guns.

In addition to these are the six battle-cruisers of the *Lexington* class. The designs for these huge ships have been greatly modified, the speed having been reduced, but the size increased. The displacement has risen to 43,500 tons, and they are thus the largest capital ships of any kind on paper or afloat. They will carry eight 16-inch and fourteen 5-inch guns.

For the purpose of comparison, Archibald Hurd considers that both battleships and battle-cruisers should be divided into two classes, since the first class, or super-dreadnought, to use a rather common but ambiguous term, are so much more powerful than the earlier dreadnoughts or battle-

ships as to require a separate designation. He considers the same thing to be true of the battle-cruisers. So, basing his comparison on the determination of the British to lay down no new ships for a while, he estimates the relative naval strength of the two leading naval powers will stand in 1924 as follows: In first-class battleships, the British have ten carrying the 15-inch gun, and we shall have ten mounting the 16-inch gun; in second-class battleships, the British have thirteen mounting the 13.5-inch gun, and our navy will have eleven, mounting the 14-inch gun.

In first-class battle-cruisers, the British have three mounting the 15-inch gun, and we shall have six mounting the 16-inch gun. Of second-class battle-cruisers, armed with the 13.5-inch gun, the British have three; we shall have no second-class battle-cruisers. Summing up, we find that by 1924 the British will have thirteen first-class capital ships against sixteen in the American navy, and that they will have sixteen second-class capital ships as against eleven in the United States Navy. As a final consideration, he asks whether, in view of the modifications which are sure to be made, as the records of the war are studied, it would not be better to delay the construction of new ships for two or three years as the British are now doing. The answer to that is that our new battleships and battle-cruisers are supposed to embody the facts of more experience.—"Scientific American."

THE INTERNAL-COMBUSTION TURBINE.

The success which has attended the application of the steam turbine to all purposes where previously the reciprocating steam engine had been employed, has naturally led engineers to consider whether the internal-combustion turbine, that is to say, the turbine in which either gas or oil are burned and the products of combustion led directly on to the rotating parts of the engine, might not be successfully employed in place of the reciprocating gas or oil engine. The internal-combustion piston engine has proved itself to be possessed of a considerably higher efficiency than the steam piston engine, and it is to be inferred that as the steam turbine is superior to the steam reciprocating engine so, too, we might expect that the gas or oil turbine would prove to be even more efficient than the internal-combustion reciprocating engine. In an article recently contributed to our columns (see "Shipbuilding and Shipping Record," April 29, p. 577) a correspondent gave some estimated figures showing the advantages which might be expected in a vessel of between 4,000 and 5,000 tons fitted with triple-expansion engines, steam turbines, Diesel engines and oil turbines, and he estimates the fuel consumption as only one-fifth of that for a reciprocating steam engine of the same power and five-sixths of that for a Diesel engine. While we do not necessarily endorse our correspondent's figures, they may be taken as indicating the ideas held by engineers regarding the superiority of the internal-combustion turbine.

It is well known that so far no ship has been fitted with an internal-combustion turbine for running under commercial conditions. But the files of the patent office reveal the fact that many attempts have been made to produce a reliable gas or oil turbine although so far these have hardly emerged from the experimental stage. It will be of interest to briefly survey what has already been accomplished in this direction and to suggest the possible directions in which improvements might be made. Probably the most successful type of internal-combustion turbine is that constructed by the Société des Turbo-moteurs, in which the Diesel cycle

is applied to a rotary engine. Air is compressed by means of a rotary compressor of the three-stage Rateau type and this is led to a combustion chamber into which the oil fuel is injected in the form of a fine spray. Owing to the heat of compression of the air, the oil immediately ignites on entering the combustion chamber, and the resulting products of combustion expand through a nozzle impinging on a turbine wheel consisting of a number of rings of blades these alternating with rings of fixed blades on the turbine casing, the burnt gases being velocity compounded as is in the Curtis type of steam turbine. A turbine developing 500 b.h.p. has been constructed on this principle.

There are two great disadvantages to this type of turbine, the first is the high temperature of the burnt gases, and the second is the loss of efficiency due to the rotary air compressor. With regard to the first of these, it must not be forgotten that the effect of gases at very high temperatures upon metallic blades depends not only upon the temperature but also upon the velocity, and at high speeds the erosive effect is particularly great. For this reason it has been found impracticable to employ blades made of some refractory substance such as quartz or slate because, while these might withstand the effects of temperature they cannot stand up against the erosive effect due to the high velocity, hence the various forms of high-speed tool steel which have been tried. Vanadium and tungsten steels have yielded good results although the life of the blades is unfortunately very short. The blades have been water-cooled similar to the pistons and cylinders of the reciprocating internal-combustion engine, but while this adds to the life of the blades, it also adds considerably to the complications of the engine itself. The problem here is obviously a metallurgical one, as Lord Fisher has remarked in referring to the development of the high-powered oil engine, and it is not unlikely that with the improvement in the heat-resisting qualities of certain kinds of steel, a metal may be found which will safely withstand the onerous conditions in the internal-combustion turbine. Attempts have been made to reduce the high temperature of the gases issuing from the nozzle by mixing them with steam, but while this may have some effect in this direction it obviously reduces the efficiency of the engine unless the latent heat of the steam can be recovered in the exhaust, and it adds to the intricacy of the machine when one of the chief advantages of the turbine is its simplicity.

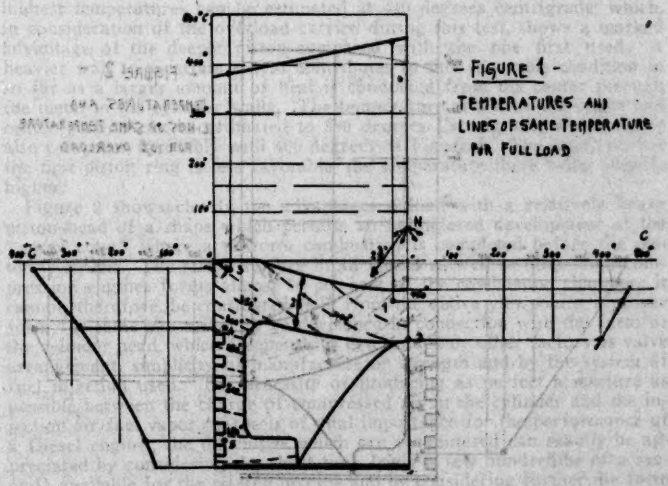
The rotary air compressor is found to be very inefficient at high pressures and yet a high pressure is essential if the temperature of the air is to be sufficient to cause ignition of the fuel spray. Low compression explosion turbines have been devised using gas for fuel, but these must be ruled out in considering a turbine for marine propulsion as a gas producer would only add to the weight and space required by the machinery. It is not unlikely that the rotary air compressor could be improved so as to become far more efficient than at present, and such an improvement would be one step towards the commercial adoption of the oil turbine. In the end therefore the problem as with the high-powered Diesel engine resolves itself into one of finding a material which can effectively withstand the effects of high temperature gases moving with high velocities. In recent years the science of metallurgy has made wonderful progress, particularly in the finding of materials of great strength. It is not too much to hope that the same patient research may ultimately yield a material which will be capable of withstanding the effects of hot gases issuing from a nozzle at high velocity.—"Shipbuilding and Shipping Record."

THE EFFECT OF PISTON DESIGN ON TEMPERATURES IN A DIESEL ENGINE.

By Paul Hentschke, M. E.

On the occasion of an extended visit to the Augsburg plant of the Augsburg-Neurnberg Engine Works, the birthplace of the Diesel engine, the writer witnessed a number of interesting tests to determine the temperatures in various sections of a Diesel engine piston. Great difficulty was experienced in obtaining reliable figures, which accounts for the narrow range of the tests. However, the subject is of such importance that no apology is necessary in bringing the fragmentary results of these tests to general knowledge in the hope that later material and more comprehensive experiments which may be in the engineering files of this country might become public.

In making these tests a vertical four-cycle engine of the standard Augsburg design was used in which the cylinder jacket and the A frame are cast in one piece. The cast iron liner and trunk piston were of ordinary design, the main dimensions being, bore of cylinder 15⅝ inches, pis-



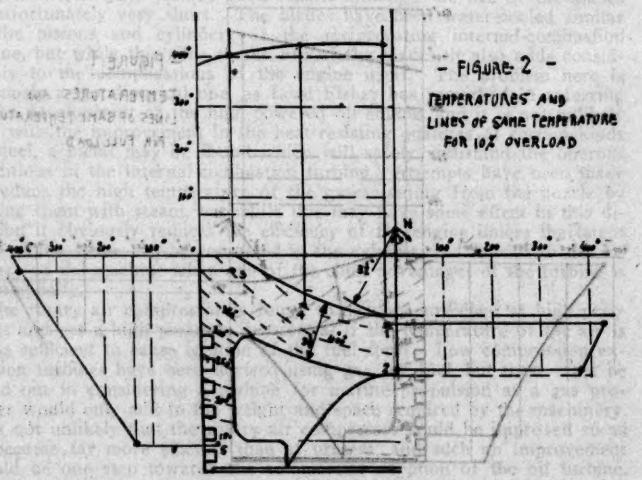
- FIGURE 1 -
TEMPERATURES AND LINES OF SAME TEMPERATURE FOR FULL LOAD

ton stroke 24 inches. The rated load of this engine was 90 horsepower and it was run at 220 revolutions per minute.

For all tests a common built-up multiple disk atomizer and a spray nozzle of 5/32-inch orifice were used. Duration of the running periods was for all tests extended long enough to ensure a temperature balance in all engine parts, and the temperature of the overflow water was maintained constant at approximately 120 degrees Fahr. for all variations of load.

The piston (Figure 1) of the test engine was originally built up, a lower body serving as a cross-head in the cylinder, while the upper part, which is exposed to the heat of combustion, carried also the grooves for the piston rings, six in number. This construction, though higher in initial cost, allows of cheaper repair, smaller spare parts and selection of a cast iron especially suitable for high temperatures. For the test reported in Figure 1, a standard piston head of slightly concave shape and about 2⅛-inch thickness was selected from stock at random; small holes were drilled from the inside in such a way that the ends of a number of thermo electric pyrometers (thermo couple) could be brought in positions as marked 1 to 5 in Figure 1. It may be observed that Couple No. 1 was approached as far as was thought advisable to the top and center portion of the piston where the highest temperature was to be expected. The hot end of Couple No. 2 was applied to the inner surface of the center portion and 3, 4 and 5 were so located on the outside of the head that a fair picture of the heat flow from the piston to the water cooled walls of the cylinder could be expected.

The relative situation of the fuel spray nozzle in regard to the position of the piston in the upper dead center is indicated in Figure 1 by Letter



N, and the smallest spraying distance from the center of the nozzle to the nearest point of the piston was 2⅜ inches. During this test an injection air pressure of 1,000 pounds a square inch was employed, and the engine was run at its rated load of 90 horsepower.

A glance at Figure 1 shows that the highest temperature, recorded at 1, is about 440 degrees Centrigrade, from which figure the maximum temperature of the piston surface may be estimated at 460 degrees, a tempera-

ture which in the dark could be observed as redhot. The lines connecting points of same temperatures, which were drawn in Figure 1 as true as the limited number of measuring points would allow, indicate an almost concentric arrangement in regard to the Nozzle N. A quick drop of temperature will be notified at the point of intersection of the side wall carrying the piston rings and the top portion of the piston head. The curves show also the temperature near the first as well as the lowest ring and the designer will do well to compare these with the coking temperatures of the customary lubricating oils.

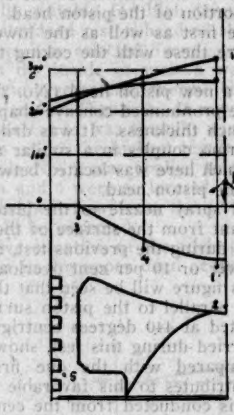
For the continuation of the tests a new piston head (No. 2) was used; it had, as shown in Figure 2, a more pronounced concave shape and also a heavier bottom portion of the 2⅜-inch thickness. It was drilled and prepared for the same number of thermo couples in a similar arrangement with the sole exception of No. 4, which here was located between Couples 1 and 3 near the upper surface of the piston head.

The relative position of the fuel spray nozzle to the piston at upper dead center is about 3½ inches distant from the surface of the latter. Injection air pressure was the same as during the previous test, the load carried on the brake was 100 horsepower or 10 per cent overload, the readings are shown in Figure 2. In this figure will be seen that the curves of same temperatures are here plainly parallel to the piston surface and the highest temperatures can be estimated at 440 degrees centrigrade, which, in consideration of the overload carried during this test, shows a marked advantage of the deeper piston compared with the one first used. A heavier wall to a certain extent contributes to this favorable condition in so far as a larger amount of heat is conducted from the center through the metal to the cylinder walls. The temperature at 2 near the inside and center portion can be estimated to 380 degrees Centrigrade, which figure also compares favorably with 400 degrees of Figure 1. The condition for the first piston ring is less favorable, the temperature there being slightly higher.

Figure 2 shows clearly the advantages gained with a relatively heavy piston head of a shape which permits an unhindered development of the "spray cone," where a uniform combustion is completed before the piston is struck. The piston surface in all Diesel as well as other high compression engines forms almost 50 per cent of the combustion chamber; it cannot, therefore, be considered only from the above viewpoints, a proper form for same has rather to be developed in connection with the form of the cylinder head, which frequently is determined by other factors as valve arrangement, simplicity of manufacture or strength and by the system of fuel injection used. The necessity of producing as perfect a mixture as possible between the charge of compressed air in the cylinder and the injection air fuel vapor charge is of vital importance for the performance of a Diesel engine; the difficulties which are encountered can readily be appreciated by considering the short time (only a few hundredths of a second) available for the mixing process and by considering further the form of the average combustion chamber, where a large portion of the charge of compressed air in the cylinder is entirely inaccessible to the direct action of the fuel spray.

A large surplus of air for the complete combustion of the fuel and therefore considerable heat losses with the exhaust gasses are the consequences. Only through extensive research work could be determined the most favorable form of fuel spray and shape of combustion chamber. More light should be thrown into the theory of the mixing process in a Diesel engine which is now understood to be effected by the violent whirls created by the blast of the injection air and the progressing combustion.

The tests with the piston, No. 2, were carried through at one-third of the rated load at various injection air pressures. In Figure 3 are given the temperatures in the Points 1, 3 and 4 at injection air pressures of 850



pounds and 560 pounds per square inch. It will be noticed that the higher pressure increased the temperature at Point 1 by about 50 degrees Centigrade.—"Pacific Marine Review, July, 1920."

THE TRANSFERENCE OF HEAT THROUGH METAL.

In marine engineering the problem of transferring heat from one side to another of a metal plate is met with in many and varied forms. The two most important are certainly in the boiler and in the condenser, in the former where the heat of combustion of the fuel has to be imparted to the water in the boiler in order to produce steam, and in the latter where the latent heat of the exhaust has to be abstracted by conduction to the cold circulating water passing through the tubes. In many respects both problems are the same, and yet in many others there are wide points of difference. It is shown in text-books on the science of heat, that the amount of heat which can be conducted through unit area of a metal plate depends upon the difference of temperature between both sides, the thickness of the plate, and upon a physical quantity termed the coefficient of conductivity of the metal. Practical experience, however, has shown that there are other factors to be taken into account, and it is only to be expected that the nature of the substance on either side of the plate, and its physical condition, although not considered in the theory of the conduction of heat, should have a vital bearing on the question of heat transference.

It has been shown by Prof. Dalby that there exists on the surface of a metal plate a thin film of the substance, whether air or water, which is capable of exerting a very great influence on the amount of heat which

passes through the plate. While metals are, generally speaking, good conductors of heat, particularly copper or copper alloys, such as are employed in the manufacture of condenser tubes and some boiler tubes, both water and air are, comparatively speaking, very poor conductors of heat, and hence the existence of the film of air or water on either side of a boiler plate or tube or of water on both sides of a condenser tube makes a vast difference to the amount of heat which is conducted through the metal. This is, of course, in addition to the further resistance to the passage of heat which is offered by foreign matter such as soot or scale, which is so readily deposited on the surfaces of a boiler or condenser. It is necessary, however, to point out that while cleanliness is a *sine qua non* of good performance, this alone is not sufficient to ensure that the maximum amount of heat will be transferred through the metal; it is also essential that the non-conducting layer of air or water shall be as far as possible removed.

Experiments in the laboratory and practical applications on shipboard have revealed the fact that one of the surest methods of increasing the amount of heat which is transferred from one fluid to another through a metal is to keep the fluids in rapid and, if possible, agitated motion. By this means, the films of fluid which adhere to the metal are more or less broken up, and a fresh quantity of the fluid is presented to the surface of the metal. Thus, for the best results in boilers, condensers and other appliances where transference of heat occurs, it is advantageous if the fluids are given a turbulent motion, so that they effectively "scrub" the surface of the metal, and in many instances this is already assured. Thus, in the ordinary marine boiler, it is recognized that the efficient circulation of the water in the boiler has a marked effect upon its steaming qualities, and many devices are upon the market which claim to impart to the water in the boiler that motion which is necessary for the effective scrubbing of the heating surfaces. In the water-tube boiler, this circulation is automatically obtained, and this is one of the reasons for the superior steaming qualities of this type of boiler. With regard to the other side of the metal, there are not many devices for imparting turbulent motion to the flue gases, although one or two can be recalled. The use of retarders in boiler tubes, twisted strips of metal which impart a screw motion to the hot gases, certainly owe their effectiveness to the fact that they lead to a more intimate contact between the gases and the surface of the interior of the tube. The use of forced or induced draught leads to an increased velocity of the products of combustion and thus to a greater heat transference through the plates and tubes, although in this case it must not be forgotten that the temperature of the flue gases themselves is higher. It will be recalled that proposals were recently put forward by a French engineer for the adoption of a system of forced or induced draught, in which the air was forced into the furnace in a series of puffs or blasts. Experiments had shown that this led to a far more efficient performance of the boiler and there can be no doubt that this is largely due to the turbulent motion which is set up in the flue gases as they pass through the furnaces and the tubes.

In the operation of condensers and similar apparatus this principle has not been so widely adopted. The motion of the water through the tubes and of the steam over them is in the nature of a steady continuous motion, although in the former case the velocity may exceed the critical velocity. We have previously expressed the opinion that the use of the reciprocating insulating pump might be found to yield better results than the use of the centrifugal pump for equal volumes of water delivered in view of the more turbulent delivery obtained from the former, and it is not unlikely that the extension of this principle to other cases where fluids

are passed through tubes either for cooling or heating purposes would be accompanied by improved results. There are, however, other factors to be taken into consideration. If the energy expended in imparting a turbulent motion to the fluid represents a greater loss than that which is gained in the increased heat transference which results, the adoption of the principle is not to be recommended. From the question of heat transference *per se*, the advantage of turbulent motion on either side of the metal is undoubted, and only by careful experiment will it be possible to ascertain whether for any particular case the advantages are not more than outweighed by the disadvantages.—"Shipbuilding and Shipping Record."

STANDARDIZATION OF SPECIAL STEELS.

Even before the war the use of special steels possessing special properties was making sure and steady progress. Engineers had got over their aversion to the practice of heat-treating steel before use because they could not deny the vast improvement that such treatment brought about, and because it gave to the steels to which it was applied characteristics which they desired and were anxious to obtain. And helped by improvements in industrial technique heat treatment had become an operation the results of which were sure and certain, and by experience it was found that an extremely wide range of tensile strength without any great sacrifice in toughness could, by heat treatment, be obtained from the different steels of special composition containing one or more alloying elements. The difficulty lay in the choice of type when for a given strength a wide range of steels were offered, and the engineer could not decide which was suited for his purpose, while generally he was not a little bewildered by the advice offered in copious measure by the numerous commercial advocates of their own brands out for a deal at any price. And so it ended, as it will generally end under similar circumstances, in the choice of the ordinary material the properties of which were already well known, so that its handling presented no difficulty even to the least skilled. This period of hesitating trial and pioneering was just beginning to be displaced by merited confidence when the war broke out and quickened to an incalculable degree the rate of progress. With the demand for guns, aero engines and tanks, the construction of which consumed large quantities of high tensile steel of special composition, grew a demand for steels of still higher strength and toughness, and this led to particularized research into the properties of new alloy steels and their heat treatment. Ideas were no sooner formed than they were tried in practice, failures were scrapped and successes were still further improved, and every day saw the demand for special steels grow greater as the applications of their use were extended. Men who would never have taken the risk of handling such materials under normal business conditions willingly stifled their doubts for the furtherance of a great cause and threw themselves energetically into the task of mastering their troubles and their lack of experience. And the result in most cases was brilliantly successful, while the permanent effect has been, and will be, of great value to our engineering industries since it has broadened their outlook and improved our knowledge of the properties of engineering materials. In five years of intensive effort there has been compressed it is difficult to say how many years of normal progress.

As the result of such rapid development it is not unnatural that some of the conclusions reached must be considered to be in the nature of tentative opinions, and to modify or confirm them time is necessary to test them under everyday conditions. At present the designer of any part of an engine can fix from his calculations what tensile strength the material should possess to bear the load with safety, but having fixed the strength of his steel he has the choice of several compositions of alloy steel, all of which can be made to possess this strength by heat treatment. The question as to which should be used then brings into view the measurement of other properties than strength and the share which such properties bear in the production of a satisfactory life of the part when in use. Should the elongation and contraction of the tensile test-piece be the sole guide in the decision? Or should the impact test, the fatigue range, the Stanton test, the bend test, the Sankey or Arnold form of alternate bend test, be used to discriminate? It is here that experimental work and experience are wanted to help to form a conclusion. There are many forms of special test available, but what particular property, if any, such tests measure is a difficult question to answer. The impact or notched-bar test has been growing in favor, but there are not a few who contend that the particular variety of toughness measured by this test and indicated by a high impact value does not give any indication of the usefulness or otherwise of the steel under examination, or bear any relation to the life in service. We believe, however, that this view is based on an erroneous idea of what the notched-bar test measures. When a localized stress of high intensity occurs in any steel part under load, local extension may occur and may lead also to the formation of small cracks which, once they are formed, may spread further and even lead to disaster. But the chances of a crack extending and the rate at which it will extend will depend on the energy required to separate the crystals without deforming the surrounding material contiguous to the surface of separation. If the crystal grains composing the material can be split or cleaved easily, then a crack once formed will require little further energy to cause it to spread. Now it is just this property of the crystal grains which the energy absorbed in a notched-bar test measures, and since it is impossible to avoid the occurrence of localized stresses it is essential to see that if they do occur the material is in the correct condition to withstand their effects. Material which has a high impact or notched-bar value—and the term high is used here in a relative sense, since it depends on the tensile strength as well—is therefore safer to use since it is much less liable to fail under irregular stress distribution which cannot be avoided in practice.

Specifications are intended to cover the average requirements and to simplify acceptance conditions, and in the case of special steels of high strength the task of the engineer would be greatly lightened if some of the numerous testing methods were done away with and the acceptance of such steels was based on a tensile and notched-bar test only. The Air Board specifications are already in existence on this basis, and their general adoption is strongly advocated.

The largest quantities of special steels are used without doubt by the automobile trade, and a very useful contribution towards the idea of standardization both in type and application is set out in a paper read before the Institution of Automobile Engineers by Dr. Hatfield. In great detail he has made lists of all the principal parts of a motor car engine and transmission system, and given for each part the type of steel which has been used and the type of steel which he recommends should be used. The information which is contained in these lists show to what a great extent individual choice affects the type of steel used for any particular

part. Take gudgeon pins, for instance. They have been made from steel with 80 tons to 90 tons tensile, from carbon case-hardening steel, plain carbon steel, 3 per cent nickel steel, nickel chrome steel, and tungsten steel. And then, on top of this, Dr. Hatfield recommends 5 per cent nickel case-hardening steel. Steering pinions have been made from nickel case-hardening steel, nickel chrome case-hardening, air-hardening steel, chrome vanadium steel and even bronze. And similar diversity exists for every part. The choice of type will depend on two factors, suitability and cost, and in general where two steels are equally suitable the cheaper steel should be used. In Dr. Hatfield's paper, however, the question of cost is not considered, and in many cases, for instance, a nickel steel is recommended where a plain carbon steel would be equally suitable and much cheaper. If a tensile strength of not greater than 45 tons per square inch is necessary, then a carbon steel will fulfill all that is wanted, and it is needless expense to use an alloy steel in such a case—always provided, of course, that strength is the main deciding factor.

But the effort to avoid confusion and to specify for each part a standardized material deserves all the support that can be given to it, not only for the automobile manufacturers' benefit, but for the steelmakers' benefit as well. There is no need to point out how standardization in types must cheapen the cost of production, and since special steels are necessarily expensive it is a further argument in its favor. It can only be carried into effect through the combined efforts of the makers and the users, and after careful consideration has been given to the conditions of service of each part and to the extensive experience gained in the use of diverse types for the same purpose. But the time is ripe for this to be undertaken, and the paper of Dr. Hatfield focussing as it does some of the differences, comes at an opportune time.—"Engineering."

TESTING TURBINE STEELS.

In determining the quality of the steel forgings which are used in the construction of steam turbines, the simple tensile test of a specimen cannot alone be taken as giving a complete index of the mechanical properties of the forging. This was clearly exemplified in the paper recently presented to the North-East Coast Institution of Engineers and Shipbuilders, by Dr. Hadfield and Mr. H. M. Duncan, in which a large number of other tests were described, all of which gave an insight into the various properties of the forging. It was admitted that the tensile test gives a good indication of the quality of the material, though perhaps not sufficient for absolute reliance. It is suggested that at least one other test might be introduced which would confirm or otherwise the conclusions drawn from the tensile test, and they recommend the Stanton test as approaching nearest to the conditions likely to be met with in practice. In this test the effects of an alternating shock are measured, blows at the rate of 96 per minute being delivered on either side of a grooved test specimen resting upon supports. The number of blows required to cause fracture at the groove is a measure of the quality of the material to resist the effects of alternating stresses which, of course, the simple tensile test does not measure. In view of the conditions which obtain in a heavy rotating mass, it is apparent that some such simple test in conjunction with the tensile test is required.—"Shipbuilding and Shipping Record."

THE ENGINEER AND NOISE.

There is no branch of science which does not come within the purview of the engineer—physics, mechanics, chemistry, for instance, are all required in measure—depending upon the actual type of plant or product with which he is concerned. Of the three branches of physics—heat, light and sound—it is generally only to the first that much attention is given, whereas it is with sound primarily that these few notes deal.

The man in charge of any type of machinery knows well the beat of his engine and the sounds which are to be expected, and experience gives a well-attuned ear for discerning any unusual sound indicating untoward circumstances or faulty operation. In the workshops noise cannot but detract from efficiency, certain noises having notably greater influence in this respect than others. It has been stated, for instance, that with certain types of marine machinery, their noisy operation leads to inefficient service of the *personnel* in charge. A number of examples could be given where noise has been an important and even a determining factor.

When gearing was first introduced in marine equipments, between the turbine and the propeller, it was thought that the sound of the gear would be a distinct disadvantage to this type of machinery, especially in passenger vessels. These fears, however, did not materialize, although, if the gearing is not well and carefully machined and aligned, it is possible for such a note to be set up as to make conditions in the engine-room—anywhere in the vicinity of the gear-case—exceedingly unpleasant, until such time as the necessary wear takes place to give the requisite tooth bearing surface for sweet running. Good running mechanical gearing gives off a "hum" which is not in any way unpleasant and which on deck or in the cabins adjacent to the engine-room can only be very faintly heard, and does not compare unfavorably with the ordinary beat of the reciprocating steam engine.

Great as the advantages of the marine Diesel oil engine undoubtedly are, silence does not generally lie to its credit. The beat of the exhaust, the clatter of the valves and the other noises attendant upon the use of such prime movers result in a relatively noisy engine-room. In time, more attention will, undoubtedly, be given to this aspect of design, and there would seem to be no intrinsic reason why the Diesel oil engine should not compare favorably in this respect with the average reciprocating steam engine. Take, for instance, automobile practice and consider the great improvements in regard to the silent operation of the engine and transmission mechanisms which have been effected within the last ten years, culminating today in a complete power plant which for silence, and refined operation, leaves little room for improvement. In this class of work, of course, there is no doubt that the introduction of the sleeve valve engine had a very distinct influence, and, but for the ensuing competition to meet the demand for a refined product, the average poppet valve engine might not have reached today the present stage of development. The "Silent Knight" engine, as it was then known, set a standard to which designers of the ordinary type of engine had necessarily to work.

Coming back to marine installations, with the average four-stroke cycle Diesel engine, what are the principal causes of noisy operation? They may briefly be stated as follows:—

1. The induction air being drawn into the main cylinders through the intake pipes.

2. The cylinder head valves, on closing, hitting their seats and, on opening, the contact of the cam with the roller.

3. The exhaust leaving the cylinder entering into the exhaust pipe, and from the exhaust pipe passing into the atmosphere.
4. The whistle of any gas or air leakage past the main piston rings.
5. The hammer action of any pumps or of their valves.
6. The natural rumble of an engine in motion.
7. The noise of the gearing driving the valve shaft, and other parts so driven.

These are given probably in the order of their importance, and they may be examined singly.

Induction may be silenced by joining up the inlets to the various cylinders by a common suction pipe and leading this pipe to a suitable part of the engine-room to ensure the relatively silent intake of dry, cool air. This system, furthermore, is to be commended as probably giving a higher efficiency of induction than individual pipes to each cylinder, and certainly has a very considerable influence in silencing this operation. The hammering of the cylinder head valves on their seats cannot entirely be avoided; there must be clearance between the roller and its cam; therefore, on closing, the valve will strike the seat with a certain velocity, and similarly on opening the cam will strike the roller. This velocity can, of course, be reduced by altering the cam shape, but it has been generally found by experience that such a course leads to a disproportionate loss of volumetric efficiency. The design, however, can ensure that the moving mass arrested when the valve strikes the seat, is a minimum, and that the weight of valve gear attached to the valve is reduced as much as possible to this end. With certain designs of valve gear, not only is the weight of the valve itself to be considered, but a certain amount of valve gear is so attached thereto that this weight must be added to the weight of the valve when calculating the mass effect. In the engine-room the exhaust from the main cylinders causes very slight noise, and can only be silenced by water-jacketting and lagging the conducting pipes. Where the exhaust passes to the atmosphere, whether in a funnel or not, the question of silence is difficult of solution, but calls for special attention, as in a number of Diesel engine driven ships this continuous beating sound is unpleasant. By means of expansion chambers in the exhaust line between the engine and the outlet to the atmosphere, and by ensuring that such noise as does take place at the exit is not intensified by structures which can give rise to "drumming:" improvement can often be made.

Piston-ring leakage should not occur to any appreciable extent, and it is desirable that the bottom sides of the cylinders should be closed in with handy doors for ready access, and that these spaces should be ventilated. It has become more general practice now to drive separately the necessary pumps, such as those for cooling water and lubricating oil, and rotary pumps which are notably silent, are gaining favor for most purposes. With a forced lubricated engine, the rumble is very small. It is caused by the displacement of air by the moving parts, and can only be reduced by having well-fitting access doors which do not rattle. Accurately-machined gearing running in an oil bath can be made remarkably silent.

If the various points to which attention has here been drawn are all considered in the design, the Diesel engine might be made to operate almost as silently as the average reciprocating steam engine. These points are raised with a view to encouraging designers to pay more attention to this subject of noise than has been the case in the past, since, from the standpoint of the operating engineer, the question is one of very considerable importance.—"Engineering."

RAPID TRIMMING OF SUBMARINES BETWEEN TWO STRATA OF WATER.

By Lieutenant Eugenio Normand, Royal Italian Navy
Submarine Section.

It is well known that in some seas the water is divided into layers of different temperature with a definite surface of cleavage between them. As the density of the water depends directly upon the temperature, it follows that the trim of a submarine when completely submerged is influenced when passing from one layer to another. In considering the possibility of keeping the submarine between two layers it is necessary to consider the two factors which must remain the same, viz., the volume of the submerged submarine and the weight of the displaced water.

EFFECT OF INCREASED PRESSURE.

The volume of the submarine may change for various reasons, the two principal ones being the pressure exerted on the hull at various depths and the changing temperature of the hull itself. Assuming that with increase of depth the form of the submarine remains absolutely the same, then the diminution of volume caused by increased pressure on the hull is of no importance, provided the hull is of circular section. In those types of submarine which have a section far from cylindrical, the pressure of the water causes a certain amount of elastic deformation, albeit a small one. To attempt to trim the vessel with the deduction of too small a volume is unwise, since the oscillations which occur while trimming increase in amplitude with the increase in the volume to be deducted, and if too small a volume has been deducted at great depths, it may lead to the vessel submerging to depths greater than the hull can safely resist. It is necessary, therefore, to ascertain the actual volume by deducing the reduction of volume caused by increased pressure from experiments made during the trials.

VARIATION OF TEMPERATURE OF SEA-WATER.

Considering next the question of variation of temperature, it must be pointed out that if the submarine is of the double-hull type, when the vessel is on the surface, the inner hull, since it is not in contact with the sea, has a temperature approximating to that of the interior which may differ considerably from that of the sea. Generally it is higher when the engines have been running any length of time either for surface propulsion or for charging the batteries, or when the vessel is exposed to the sun. On submerging, when the ballast tanks are flooded, the inner hull becomes cooled and has a tendency to twist. The cooling proceeds slowly, however, since it is compensated by the heat generated in the electric motors and batteries, and by the life of the crew in the submarine. It is only some time after the vessel commences to dive that any difference of temperature can be observed. The volume of the hull changes slowly on this account but only a few cubic millimeters for every 100 tons of displacement and for every degree of reduction of temperature. These variations are, therefore, of no great importance.

CHANGES OF DENSITY.

The density of the sea-water also changes for several reasons, among which may be mentioned increase of pressure, variation in the quantity of salts in solution and changes of temperature. The first of these is so small that in practice it may be neglected, but the second and third have

considerable importance in studying the possibility of immediate trimming of submarines. The effect of salts in solution is to cause a considerable increase of density which, measured at a standard temperature of 17.5 degrees C., varies between 1.010 in the Baltic Sea to 1.040 in the Red Sea, and assuming that a submarine may operate in waters where the density varies, say, from 1.010 to 1.030, it is obviously necessary for the volume of the compensating tank of the vessel to be not less than 2 per cent of the displacement when in surface trim. For example, a submarine of 250 tons, passing from the lower to the upper Adriatic Sea, meets with water changing from 1.0294 to 1.0252, so that the change of buoyancy is equivalent to a loss of one ton in displacement, this being due to the change of four-thousandths in density. Similarly, in passing from the Mediterranean Sea to the Black Sea there is a change of density of fifteen-thousandths, representing a change of buoyancy of nearly four tons. Again, the composition of the water varies at different depths, particularly in the Mediterranean Sea, and in zones near large rivers where the mixing of fresh and salt waters is found to result in distinct layers of water of different density depending upon the seasons, whether the sea is calm or rough, and the dominating currents.

The effect of variation of temperature is even more pronounced. The coefficient of increase of volume varies with the temperature as shown by the following table:—

From 10° C. to 15° C.	0.00015
From 15° C. to 20° C.	0.00022
From 20° C. to 25° C.	0.00027
From 25° C. to 30° C.	0.00033

CALM OR ROUGH SEA EFFECTS.

In the Mediterranean Sea the temperature in the summer changes from 25 degrees C. at the surface to 15 degrees C. at a depth of 25 meters, and since the increase of buoyancy for every degree of fall of temperature is 0.24 kilogram per ton, for a submarine of 300 tons displacement in surface trim there is an increase of buoyancy of 72 kilograms per degree or 720 kilograms, in submerging to a depth of 25 meters. In the Adriatic Sea, the author has observed a fall of temperature of one degree for every two meters of depth from the surface, so that it is necessary with a 300-ton submarine to add 35 kilograms of water for every meter that the vessel dives. This effect is less marked when the sea is rough. According to the roughness of the sea, a surface zone acquires a uniform temperature, but when this is passed through the temperature falls rapidly. In perfectly calm weather, but after a period of rough seas, zones of 10 and 17 meters in thickness, with temperatures of 23 degrees C. and 19 degrees C. were observed, and after a further increase of depth of one meter the temperature fell to 16 degrees C., this constituting a layer of water in order to pierce which an increase of 200 kilograms was required in a vessel of 300 tons displacement. These layers are particularly noticeable in spring and summer, but in autumn they become less noticeable and in winter they almost disappear.

EXACT TRIMMING.

In conclusion, it should be remarked that the change of buoyancy due to temperature must always be considered if it is desired to effect rapid trimming of the submarine. This can be obtained by measuring the temperature of the sea at different depths by means of a thermometer inserted through the side of the conning-tower, the vessel being maneuvered

in depth until a zone of water is reached where the changes of temperature are not very sudden. In such a zone the trimming of the vessel will be simpler and more lasting. It has been found that by making allowance for the variation of temperature, particularly in spring and summer, and when the sea is calm or has recently been so, that it is possible to so trim a submarine that she floats with the periscope scarcely showing above the surface, it appearing as if the submarine remained suspended by the periscope. Then, if the periscope is drawn in, it is possible to remain at this depth totally submerged, but without diving. The author has himself remained on several occasions resting for periods of eight to twelve hours at depths of 15 to 20 meters, although at these depths the periscope cannot appear above the surface.—"Shipbuilding and Shipping Record."

ALIGNMENT METER FOR TURBINE GEARING.

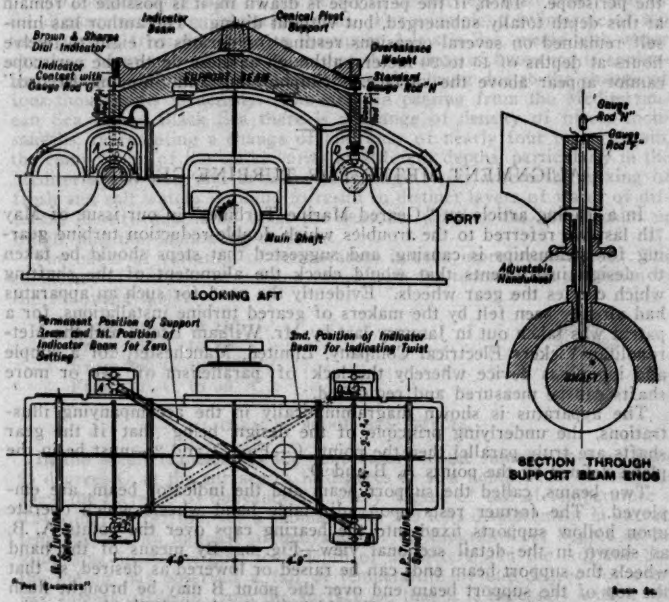
In a leading article on "Geared Marine Turbines" in our issue of May 7th last we referred to the troubles which double-reduction turbine gearing for steamships is causing, and suggested that steps should be taken to design instruments that would check the alignment of the shafting which carries the gear wheels. Evidently the need for such an apparatus had already been felt by the makers of geared turbine installations, for a patent was taken out in January last by Mr. William Eccles and the Metropolitan-Vickers Electrical Company, Limited, Manchester, for a simple and ingenious device whereby the lack of parallelism of two or more shafts can be measured and registered.

The apparatus is shown diagrammatically in the accompanying illustrations, the underlying principle of the design being that if the gear shafts are truly parallel then the point C—Figs. 1 and 2—must be in the plane defined by the points A, B and D.

Two beams, called the support beam and the indicator beam, are employed. The former rests upon adjustable hand wheels which operate upon hollow supports fixed into the bearing caps over the points A, B, as shown in the detail sectional view—Fig. 3. By means of the hand wheels the support beam ends can be raised or lowered as desired, so that the top of the support beam end over the point B may be brought flush with the top of the gauge rod F, which passes through the hole in the end and rests on the top of the shaft at B. Similarly the gauge rod E—Fig. 1—fixes the vertical position of the end of the support beam above A, and in this manner the position of the conical pivot on top of the support beam is provisionally fixed for the purpose of the test relatively to the points A and B on the shafts. The gauge rods E and F are now taken out and replaced by gauge rods G, H respectively. The indicator beam is next placed on the conical pivot support so that the faced underside of the overbalance weight—see Fig. 1—rests on the top end of the gauge rod H. It will thus be evident that the indicator beam rests on the pivot support and the gauge rod H. At the other end of the indicator beam there is a Brown and Sharpe dial indicator graduated in thousandths of an inch. The contact knob of this dial indicator rests on the top of the gauge rod G, and in this way a reading is obtained. The dial is turned to make this reading zero.

The indicator beam is now swung round, so that the overbalance weight is above D, the facing under the weight resting upon the same gauge rod H, which has been placed in this second position. The indicator contact knob rests on the gauge rod G in its new position C, and if the indicator

shows a zero reading the two shafts A C and D B are in the same plane. If a reading other than zero is obtained, then the plus or minus register will serve as a guide for chocking up the gear case. Incidentally it may be remarked that for testing for co-planar alignment in the works the support beam might be dispensed with and the pivot support carried directly by the gear case top.



FIGS. 1, 2, AND 3—ALIGNMENT METER FOR TURBINE GEAR BRAKES

All the adjustments and settings described above are made when there is no load on the gearing, and when it is not in motion. When the gearing is running, in view of the fact that there is always a certain amount of clearance in the bearings, it is probable that the shaft journals will rise or fall in their bearings, and this condition would effect the zero setting above referred to. To obviate this trouble the gauge rods E and F are replaced through the support rod ends over A and B respectively, and the supporting hand wheels are readjusted to bring the tops of the gauge rods flush as before. In this way the conical pivot support is brought back to the same position relatively to the points A and B, and the zero setting regained, while the indicator beam is still in the working position over C and D. With these points attended to the dial indicator may be read at any time. It should, of course, be understood that the diameters of the journals at A, C, D and B should be very carefully measured by a micrometer gauge, and when they are known, the necessary corrections of the readings can be made for the relative positions of the shaft center

lines. The meter is shown applied to the high-speed pinions only, the assumption being that these pinions were in line when erected in the shops, and when all the necessary bearing adjustments required to make the low, intermediate, and high-speed shafts line up were made. In such circumstances, when the gearing is erected on board ship, it is only necessary to get the high-speed pinions true with each other, when the running shafts will also be in line, if the gear case is strong enough to resist cross

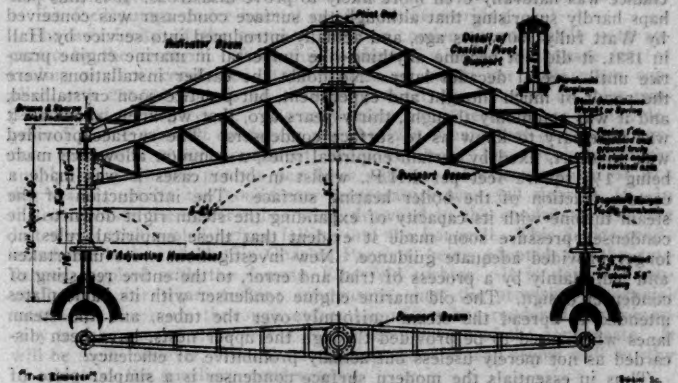


FIG. 6.—SUPPORT AND INDICATOR BEAMS FOR ALIGNMENT METER.

bending. If, however, it be possible for the gear casing to spring or bend transversely, then it would be necessary to apply the meter between the low-speed shaft and each pinion.

It will be seen from Fig. 4 that the beams are of very light construction, and are built up of some light metal such as "duralumin." The weight of the indicator beam, if made of this metal, would be in the neighborhood of 25 pounds, and that of the support beam 35 pounds, so that the total load on each hand wheel would be about 30 pounds.—"The Engineer."

SURFACE CONDENSER PROBLEMS.

The training of the sea-going engineer of fifty years or more ago comprised careful instruction in the precautions necessary to prevent the deposit of salt in the boilers, and he was provided with and used regularly instruments by which the salt content of the boiler water could be determined. He was, moreover, warned that the boilers must be blown down regularly every few hours and filled up again with fresh sea water if serious trouble was to be avoided. The waste of heat in this process was of course obvious, but nevertheless, it took many years to introduce the surface condenser into sea service. Marine superintendents have, however, always been highly conservative, and much more inclined to accept the evils they know and can more or less control, than to risk the

possibility of unforeseen difficulties arising from the working of apparatus previously untried at sea. For this lack of enterprise they are perhaps hardly to be blamed. Neptune, according to Kipling, exhorted the pioneer navigator to hesitate before he tempted the seas with untested expedients, and warned the adventurer that he must not "make at all, or all make good" his bulwarks and his boasts. A breakdown at sea is still a very serious matter even when, as now, the traffic is pretty dense on most of the ocean highways; and in the middle of last century such a mischance was naturally even more likely to prove disastrous. It is thus perhaps hardly surprising that although the surface condenser was conceived by Watt fully 150 years ago, and actually introduced into service by Hall in 1831, it did not become anything like universal in marine engine practice until several decades later. No doubt the earlier installations were the result of much thought and experiment, but practice soon crystallized, and it was commonly thought, thirty years ago, that we knew about all it was necessary to know as to surface condensers. The surface provided was, in short, fixed by certain empirical rules, a common allowance made being 1½ square feet per I.H.P., whilst in other cases it was made a certain fraction of the boiler heating surface. The introduction of the steam turbine with its capacity of expanding the steam right down to the condenser pressure soon made it evident that these empirical rules no longer provided adequate guidance. New investigations were undertaken and led, mainly by a process of trial and error, to the entire recasting of condenser design. The old marine engine condenser with its baffle plates intended to spread the steam uniformly over the tubes, and the steam lanes which used to be provided through the upper nests, have been discarded as not merely useless but actualy prohibitive of efficiency.

Thus in essentials the modern surface condenser is a simpler piece of apparatus than the type it has superseded. Designers have gradually been forced to realize that in the main condenser efficiency was dependent on the reduction to a minimum of the amount of air collected in the casing, and it further became apparent that such an irreducible minimum of air as had perforce to be retained must be prevented from jacketting the tubes. To this end the steamway is contracted systematically as condensation proceeds, so as to maintain to the ultimate end a good rush of steam over the tubes.

Condensers are, however, built for other purposes than for restoring to liquidity the steam exhausted from a steam turbine, and in these condensers, types of design discarded as injurious where a high vacuum is desired, are retained, and in some cases it would seem with considerable advantage.

In a paper* read on Monday evening last, Mr. C. Waldie Cairns and Dr. J. Morrow described a series of experiments on condensers, which they have carried out on behalf of the North-East Coast Institution of Engineers and Shipbuilders. The condensers which were used for the tests were of the type employed for condensing the steam from the auxiliary machinery on shipboard. Here the steam is condensed at atmospheric pressure and is led directly to the boiler-feed pumps. Four types of condenser were tested, of which two of the "contraflo" design were provided with departmental drainage. One condenser was of the common cylindrical type, and the fourth of the pear-shaped pattern now so largely adopted in steam turbine practice. The smallest condenser

* Report to the Council of the North-East Coast Institution of Engineers and Shipbuilders on Tests of Auxiliary Condensers, by C. Waldie Cairns, M.Sc., and J. Morrow, M.Sc., D.Eng.

experimented with had 250 square feet of surface, and was intended to condense 8,750 pounds of steam per hour, whilst the cylindrical condenser, with 600 square feet of cooling surface, was rated only as good for 7,500 pounds of steam per hour. This also was the rated capacity of the pear-shaped condenser, which had 350 square feet of cooling surface. In the tests the actual condensing capacity was fixed by determining the quantity of steam which, with the supply of circulating water available, could be condensed without the temperature of the condensate rising above 180 degrees F., which was considered to be the limit for the supply of the feed pumps under normal conditions of working.

The inferiority of the common circular condenser was made very apparent by the research. Thus this condenser, with its 600 square feet of cooling surface, condensed 11,200 pounds of steam per hour when supplied with 235,000 pounds of circulating water per hour at an inlet temperature of 91 degrees F. The condensate had a temperature of 141 degrees F. The smallest condenser with but 250 square feet of cooling surface condensed 10,000 pounds of steam per hour, the circulating water supply being 148,200 pounds per hour with an inlet temperature of 89.5 degrees F., whilst the condensate temperature was 143 degrees F. In this condenser the lower tubes were drowned in the condensate and the latter was thus cooled before falling into the hot well. This device originated, we believe, with Sir Charles Parsons, being adopted for the condenser supplied with the famous Elberfeld turbine. It is not now generally used for steam turbine condensers, since the provision of special means for the more thorough extraction of air has rendered it unnecessary to take special steps to reduce the temperature of the condensate. It will be seen, however, from the foregoing that the plan seems well worthy of adoption for auxiliary condensers.

In their introductory observations, Messrs. Cairns and Morrow remark that the apparatus at their disposal was such as to make possible merely a comparison of the performances of the four condensers submitted to test, and that they were unable to extend the investigation into one on the laws of heat transfer.

Of recent years a good many experiments directed to this end have been made both here and in America, and the subject has also been investigated theoretically. In no case can it be said that much progress has been achieved. Too many experiments have failed to note that the resistance to the transfer of heat from steam to circulating water is complex and cannot be expressed by a single term. On the steam side, the resistance is immensely increased by the presence of air. As the steam drifts towards a cold tube it carries with it any air entrained. On reaching the cooling surface the steam disappears as water, but the air remains jacketting the tube and hindering the access of fresh steam. It is this that makes it essential for securing good results at high vacua, to maintain a rush of steam past the tubing so as to sweep away the air which tends to collect there. Apart from this air jacket practically the whole resistance to the heat transfer on the steam side is to be found in the film of water deposited on the condensing surface. This seems to have an average thickness of the order of one mil. and is of course invisible. As it thickens, however, the excess collects up into drops, which can be seen on the surface of the tube if means be provided to this end. The resistance to heat flow through the tube itself is very small, but that on the water side may be very considerable even in the absence of the deposit to which condenser tubes are so liable under certain working conditions. With suitable provision for the extraction of air the resistance to heat flow on the steam side of the tubes is probably fairly constant, but this

is far from being the case with the resistance experienced on the water side even with perfectly clean tubes. Quite commonly experimenters, both here and in America, have measured the total resistance to heat transfer from the steam to the water and endeavored to find a formula for the heat flow, considering this total resistance as consisting of one term only. Some very curious formulas have been proposed accordingly, but the trouble that has invariably arisen is that each new series of experiments has required a new formula to express the results.

On the theoretical side the pioneer was Osborne Reynolds, but his work was not only incomplete but has been frequently misunderstood and misapplied. Many have thought it possible to deduce from his formula the wall temperature of the tube, but what he really deduced was not this, but a formula giving the temperature of the inside face of the film of liquid which adheres to the walls of the tube and on the average takes no part in the general turbulent flow of the mass of the circulating fluid. Even so, his formula is only applicable to very low rates of heat transfer.

In a paper read some eighteen months ago before the Institution of Engineers and Shipbuilders in Scotland Professor A. W. Porter, F. R. S., attempted to deduce from the theory of dimensions the thickness of the viscous film which slides along the wall of the tube, and the resistance of which constitutes undoubtedly a large fraction of the total resistance to heat transfer. Here again, however, the method employed was valid only if applied to cases in which the rate of heat transfer is extremely small, a condition which was not satisfied in Webster's experiments, the results of which were employed by Dr. Porter to determine the value of his constants.

In fact, in both Reynolds' formula and in that of Dr. Porter there is a term dependent on the viscosity of the water, but this viscosity in condenser practice is very far from being constant across any given section of the tube. Thus the fact that the average temperature, viscosity and velocity of the water is the same in one tube as it is in another of the same diameter is no guarantee whatever that the resistance to heat transfer will be the same in the two cases. In fact, the higher the temperature of the tube the smaller will be the viscosity of the adherent film of fluid, and this film will thus be thinner and offer less resistance, the steeper the temperature gradient along the cooling tube. It would appear, therefore, that any rational formula for the resistance to heat flow on the water side of a condenser tube must include a term dependent on the temperature gradient along the tube and cannot be expressed in terms merely of the average velocity and average viscosity of the water and of the diameter of the tube.

In the meantime engineers have to make condensers and to do the best they can without the aid of a really rational theory of heat transfer. Such a condition of affairs is no novelty, and it is notorious that the steam engine was brought to a high pitch of perfection, with but little guidance save of the most general character from thermodynamic theory, and probably the best present practice in condenser construction would benefit but little by further development of its theory. At the same time a similar feeling to that which makes us take an interest in facts about distant suns like Sirius, or in the details of the ancient civilizations of Egypt and Mesopotamia, impels the engineer to desire fuller reasons for his methods of work than the mere fact that such methods have been successful in the past. Rules of thumb thus provided are often exceedingly convenient, but the designer can never feel certain as to how far he is justified in trusting to them outside of the limits actually covered by experience in the past. This in itself is sufficient reason for the desire

to rationalize all engineering formula, and for further experimental and theoretical inquiry into the laws of heat transfer. Such, however, to prove useful must be made with a fuller appreciation of the real nature of the problem than has commonly been the case in the past. A clear distinction must be made between the resistance to flow on the steam and on the water sides of the tubes, and each determined independently and in conditions approximating fairly to those ruling in actual condenser practice.—"Engineering."

REDUCTION GEARING ON STEAMSHIPS.

The phenomenal success of the earlier installations of reduction gearing on turbine-driven steamships has of late met with a check owing to the fact that certain gears recently fitted have suffered excessive wear in service. It is noteworthy, however, that the failures are confined to double reduction gears, whilst single reduction gears still continue to give satisfaction in spite of large increases in the load imposed on the teeth and in the pitch line speed. In one case in which 18,500 horsepower is transmitted through a single pinion the pitch-line speed is 132 feet a second and the load per linear inch of pinion face is over 1,000 pounds. In another installation this load is 1,145 pounds per linear inch, and the pitch-line speed is 125 feet a second. Breakages have been astonishingly few, and where they have occurred have been traced to faults in the material or in the minor details of the design. In his recent valuable contribution to the proceedings of the Institution of Naval Architects, Commander Tostevin stated that out of a total of 556 sets of single reduction gears supplied to the Navy only three have had to be removed for refit, and this necessity arose not from any failure of the gearing proper but to the fact that owing to the straining of the hull of the ships during service, the turbine and gear shafts had got a little out of alignment. There have been only two cases of broken teeth in service, but on removal of the broken or cracked material the gears have continued to run satisfactorily.

In the mercantile marine single reduction gears have proved equally satisfactory. Merchant ships however, for the most part are low-speed vessels, and it is inconvenient to obtain with this system as high a gear ratio as is desirable if both turbines and propellers are to be driven at their respective best speeds. This has led to the adoption of double reduction gearing which in certain instances has proved quite satisfactory in service, but in other cases the wear has been excessive. These experiences have led to a demand for a decrease in the loads imposed on the teeth, but it is exceedingly questionable if such a policy is in any way well based. In fact, there is strong reason for believing that the loads now carried could be substantially increased, provided certain modifications were made in the design of the installation as a whole. In short, whilst the proposed reduction in the loads now imposed on the teeth would probably improve matters, it seems practically certain that the benefit would in the main be due merely to the fact that the weight of the last motion wheel would be increased. There are, however, cheaper methods of securing flywheel effect than that of merely increasing the diameter or length of this wheel.

The trouble with double reduction gears appears to be confined to the second motion or slow-speed pinion, which in some cases has worn seri-

ously during a comparatively short period of service. The high speed or first-motion pinion, designed to the same constants, has shown as little wear as in single reduction gearing.

This feature has led several engineers to the probably well-founded belief that it is the variation in the propeller torque which is in the main responsible for the trouble. In the December issue of International Marine Engineering, Mr. W. L. Emmet states that gears have failed at sea under loads less than one-quarter of what similar gears have, in land service, carried without trouble. He mentioned incidentally that with an experimental gear tested on land by the General Electric Company, a load of 3,000 pounds per square inch was borne without trouble, the pitch-line speed being 116.6 feet per second. As indicative of the extraordinary variations which may arise in propeller shaft torque he gave the diagram which we reproduce in Fig. 1. This was recorded by a

torsionmeter on a ship steaming in ballast through a fairly rough sea. Whether this can be considered a fairly representative case or not, it at least does show that the propeller torque may be expected to vary cyclically through a large range. With a single reduction gear the wheel on the propeller shaft is necessarily large and heavy, and its inertia must greatly reduce the shocks transmitted to the teeth. A single shock even of extreme violence does it would seem little damage. In a recent discussion at the North-East Coast Institute of Engineers and Shipbuilders, Mr. J. C. Blackburn stated that whilst steaming at full speed the *King Orry* passed over a submerged rock. In doing so she lost all the blades of one propeller, and the propeller shaft was bent so much that it jammed. Nevertheless the gearing escaped unscathed.

On the other hand there is some very conclusive evidence that high speed gearing does not stand up well to large and repeated fluctuations in the load, such as might arise if the cyclical fluctuations in the propeller torque reached the teeth of the wheels undamped by the inertia of heavy revolving parts.

In our issue of October 24 last, page 555, we gave particulars of a case in which de Laval gearing (which is in general a very reliable piece of mechanism) gave way repeatedly under conditions which were apparently in no way abnormal. The gear transmitted 425 horsepower to an air compressor and failed repeatedly in service, having to be replaced no less than five times in six years. Ultimately it was discovered that the actual variations in the compressor torque were very materially higher than had been supposed, and when the compressor in question was replaced by a Reavell machine requiring a nearly uniform torque under all conditions of service, the trouble ceased. It is noteworthy that in this instance the final drive was taken through a belt. This would act very much in the same way as an elastic coupling, yet the teeth failed as stated.

This experience affords strong confirmation of the view that the at times unsatisfactory behavior of double reduction gears is due to insufficient inertia at the last motion or propeller shaft. In the early '80s the Holt Line of steamers were commonly provided with single-crank engines, having a large flywheel mounted on the propeller shaft. Marine engineers who ran these engines declared that their behavior at sea was exceptionally satisfactory.

It will be an exceedingly interesting case of history repeating itself should present troubles lead to a reintroduction of this or an equivalent expedient on ships fitted with double reduction gearing. Certainly such a reversion would be less costly than the alternative advocated in some quarters of reducing the nominal load on the teeth.

The fact that the trouble experienced has been in the nature of excessive wear, and not lack of strength, raises anew the question as to the rational relationship between the diameter of a gear wheel and the permissible load on the teeth. American engineers have held from the outset that the load imposed should vary in direct proportion to the diameter of the pitch circle. In this country the load per inch run of pinion face has commonly been calculated from the equation:

$$W = C\sqrt{d} \text{ (lb. per inch run).}$$

Here W denotes the load in pounds per linear inch, d the diameter of the pinion in inches and C a coefficient, which originally fixed at 170 has, in at least one successful installation, been as high as 308. In his discussion on Mr. Walker's paper, read last December before the North-East Coast Institute of Engineers and Shipbuilders, Mr. S. S. Cook attributed the adoption of this rule to a conservative desire to err, if at all, on the side of safety, when fixing the proportions of large gears on the basis of experience with small installations.

During the last couple of years we have learnt a great deal about lubrication which was previously unknown, and these later researches go far to confirm the American view (based, however on entirely different considerations) that the load imposed on teeth should vary directly as the diameter of the pitch circle and not in proportion to the square root of this diameter. The crux of the question depends upon whether there is or is not a minimum limit to the thickness of the lubricating film. For a given maximum intensity of pressure on the teeth the minimum thickness of the oil film does vary nearly in proportion to the square root of the pinion diameter. Recent work, however, indicates that for practical purposes there is no minimum limit to film thickness, as it has been proved that lubrication is possible with a film of which the thickness is a few molecules only, that is to say with a film of which the thickness is but a small fraction of one-millionth of an inch. Failure of gear teeth for other reasons than the breakdown of the lubrication would occur long before the film of oil between the teeth was reduced to such a thickness as this.

Nevertheless, even if as our present knowledge suggests, the square root rule is irrational, its adoption has probably been fortunate, as the importance of inertia in the propeller shaft wheel was for the most part unsuspected. The very moderate loading of large gears which has been a consequence of this rule, has resulted in the use of larger wheels on the propeller shaft than would have otherwise been provided, and the gain in inertia has apparently in the case of single reduction gearing been sufficient to prevent the teeth of this wheel, "overrunning," to any serious degree, the pinion teeth, as a consequence of the fluctuation in the propeller torque.

Whilst transmitting torque the teeth of the gear are under strain, and potential energy is stored up in them and in other parts of the gear just as it is when a spring is compressed. If the torque on the propeller shaft vanishes this potential energy becomes available for accelerating the last wheel of the gear, and if this be thereby flicked round faster than the driving wheels of the gear can follow up, the teeth will separate and will suffer a hammer blow on again coming into engagement. The stress developed on the concussion may be very high, and in excess of the elastic limit of the material. According to experiments made on mild steel rollers by Mr. J. Christie (Trans. Am. Soc. C. E., vol. xxxii) this limit is attained in static tests when the load taken in pounds per inch of "line contact" exceeds 1,600 R, where R is the radius of curvature of the roller taken in inches. In these experiments there was no effective lubrication, and in view of the effect of an oil film in distributing the pressure, gear teeth should be capable of carrying a somewhat greater load than the above before the elastic limit of the metal is exceeded. The mathematical theory of the behavior of a lubricant when the teeth "hammer" leads to one result which may be put in a paradoxical form, viz.: In all conditions an infinite time must elapse before the metal surfaces of the teeth come into actual contact. Another and less startling statement of this result is, that it is impossible to squeeze out from between the surfaces the last molecules of the lubricant. Thus expressed, the conclusion seems not unreasonable, even though the assumptions on which the above mathematical deduction is based, fail to hold up to the bitter end.

As already noted experience indicates that a single concussion, even if very violent, will do little harm to the teeth of a gear, but under repeated blows the metal may shell off, and this is exactly what is found to occur. It is noteworthy, as confirming the view above put forward, that the teeth of marine reduction gears often suffer from pitting in the neighborhood of the pitch point. This is exactly what might be expected. Since the deflection of a cantilever under load is proportional to the cube of the length, the stiffness of two teeth, considered as an elastic system, will be a maximum when the point of contact is near mid height, and will be much reduced when the point of one tooth bears near the root of the other. Hence, when hammering occurs the teeth in contact near the pitch point have to take far more than their fair share of the shock. It is noteworthy that some gears in which pitting has occured continue to run satisfactorily. Presumably the process has in these cases worn down the surface near the pitch point, sufficiently to ensure that a tooth in engagement near its point shall take a fair share of the shocks. Pitting would seem to arise from the hammering of the teeth, and to indicate that it would be advantageous to increase the rotary inertia of the propeller shaft or last wheel. Where pitting is absent another form of wear has been observed. The metal of the teeth becomes plastic under the pressure where this may be locally excessive and flows. The lubrication is, however, fully maintained, and for all practical purposes the teeth are uninjured. Plastic flow of this kind is probably largely responsible for the success of rigidly mounted gears. The teeth are able to wear themselves to a uniform bearing at full load without failure of the lubrication.

Whilst the addition of weight to the propeller shaft wheel appears to offer one solution of the problem of excessive wear, another and, if practicable, preferable method of reducing the risk of "overrunning" would be to diminish the angular momentum of the turbine rotor. It seems probable that one type of turbine may be more suitable for driving reduction gearing than another, and it is perhaps not without significance that the loads imposed on the teeth of de Laval gears are remarkably low.

Thus, in the case of a de Laval turbine of 300 horsepower, the pinion is 2.504 inches in diameter and the load carried is 64.8 pounds per inch of pinion face. The stress at the root of the teeth due to bending is but 1,165 pounds per square inch. In most marine reduction gears, the corresponding figure ranges from 4,000 pounds to 6,000 pounds per square inch, and in one installation it is as much as 8,500 pounds per square inch. In this case the pinion is 10.7 inches in diameter, the pitch-line speed 125 feet a second, and the load carried per inch width of pinion face 820 pounds. The wide discrepancy between these figures and those adopted in proportioning de Laval gears certainly requires accounting for, and the most obvious explanation is to attribute it to the high angular momentum of the turbine wheel. This wheel cannot readily change its speed, and thus if large fluctuations occur in the torque on the driven shafting the teeth will "overrun" each other and be destroyed by the consequent mutual cold hammering.

It would be of great interest to know whether gear troubles have been most common with reaction or with impulse turbines. If there is any difference in this regard it will probably be found that the angular momentum of the one type of turbine rotor is higher than that of the other.

The mathematical theory of the lubrication of gear teeth was worked out in an article published in our issue of August 16, 1916. This theory, it may be observed, is more directly applicable to gear teeth than to any other case of lubrication met with in practice. The effective width of the film under pressure is small in comparison with its length, and end flow of the oil is accordingly negligible. The teeth, moreover, come into gear intermittently. They have therefore time to cool between each successive engagement and the heat generated in the film is conducted away at each engagement by relatively cool metal surfaces. The viscosity of the oil should therefore remain more nearly constant than it does in other cases to which Reynolds' theory has been applied. Hence the calculated results of the theory should be directly applicable to practice with relatively little correction.

The relative motion of the two teeth in engagement with each other is in the main one of rolling, and is thus analogous to the motion of a cart wheel over a muddy road. The conditions accordingly differ greatly from those pertaining to journal lubrication. In the latter case the shearing stress in the film is a maximum at the point where the film is thinnest. This is not always true when the relative motion of the opposing surfaces is one of rolling, since although, mathematically speaking, the shear does attain a maximum at this point, there is a second and in some cases at least, a higher maximum elsewhere.

Again, in starting up a journal bearing from rest the layer of oil between the opposing surfaces does not act as a whole until a certain speed is attained. It is simplest to consider the case of a bush completely surrounding the journal. With the journal at rest, the point of nearest approach of journal and brass is immediately in the line of the load, as indicated in Fig. 2. In the stable running position, however, with the whole interspace filled with oil, all taking an effective part in the lubrication, the point of nearest approach is at right angles to the line of the load as represented in Fig. 3. Hence on starting up such a journal bearing in order to attain the position at which the film is operative as a whole it is necessary to lift the load. At the outset only a narrow band of the film directly under the load is effective for lubrication. When the load begins to lift, the then effective part of the oil film is thickened and loses its supporting power, and consequently the load falls back again, to be relifted by the greater pressure thereby developed in the film. This sort

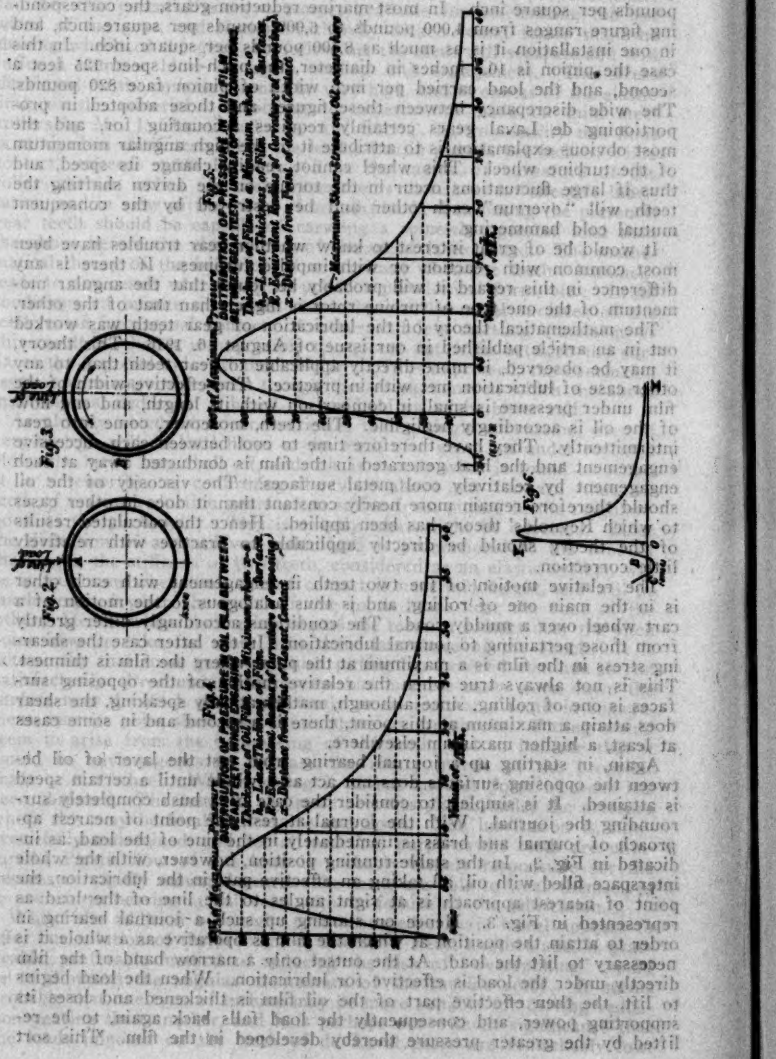


Fig. 3
Fig. 4
Fig. 5
Fig. 6

of alternation goes on until the speed attains a certain critical value at which the whole of the oil present in the interspace begins to act as a unit. The load is then permanently supported, and the point of nearest approach of journal and bush is at right angles to the line of the load as indicated in Fig. 3.

Nothing in any way analogous to this occurs in the case of teeth entering into gear. They approach each other both covered with a film of oil, which is then squeezed out as the two surfaces roll over each other. This squeezing out is resisted by the viscosity of the lubricant.

In the article already cited *supra* it was shown that the curve representing the pressure in the film of oil might have the form represented in Fig. 4. This curve corresponds to the case in which the teeth are in the act of engaging. When disengaging the conditions are less favorable, and unless the tips of the teeth are rounded off the maintenance of the lubrication will be dependent here on the relative sliding of the teeth and not on their relative roll. When fully engaged the pressure curve may be of a more effective type, and under the best conditions may have the form represented in Fig. 5. In this case the load carried is a maximum for a stated value of h_o, which represents the minimum thickness of the oil film. Another possible form of the pressure curve is indicated in Fig. 6. In this case it will be seen there is a negative loop on the "trailing" side of the point of closest contact.

The equation to the curve represented in Fig. 4 (all quantities being taken in C.G.S. units) is:

$$P = \frac{\lambda u_o}{h_o^2} \cdot \sqrt{2 R h_o}\,(\sin 2\alpha + \tfrac{1}{4} \sin 4\alpha)$$

Here P denotes the pressure in dynes per square centimeter, λ the absolute viscosity of the oil in C.G.S. units, and h_o the least thickness of the oil film, in centimeters. The velocity with which one surface rolls over the other is represented by u_o, and is expressed in centimeters per second. For involute gearing this rolling velocity is approximately one-third of the pitch-line speed. R denotes the equivalent mean radius of curvature of the teeth, expressed in centimeters. For involute teeth the value of R may be taken as given by the following equation:

$$\frac{1}{2R} = \left[\frac{1}{D_0} + \frac{1}{D_1}\right] \frac{\cos\phi}{\sin\theta}$$

Here D_0 is the diameter of the pitch circle of the pinion and D_1 the diameter of the wheel, whilst θ denotes the angle of obliquity or pressure angle of the teeth and ϕ the angle the helix makes with the axis of the wheel. With modern gears in which the angle of the helix, to which the teeth are cut, is 30 degrees, the value of θ is about 16½ degrees.

Finally, α is defined by the equation:

$$\tan\alpha = \frac{x}{\sqrt{R h_o}}$$

where x denotes the distance in centimeters of any point in the film from the point of closest contact.

When the distribution of the pressure is that represented in Fig. 5, its value is given by the expression:

$$P = \frac{\lambda u_o}{h_o^2}\sqrt{2 R h_o}\left[0.4841\left(\frac{\pi}{2} - \alpha\right) + 0.6773 \sin 2\alpha - 0.4597 \sin 4\alpha\right]$$

As already stated, in this case the load carried for a given value of h_o is a maximum. This distribution of pressure is not possible when the point

of contact is at the end of a tooth. Owing, however, to the fact that the teeth are elastic, the load imposed on the teeth is greater when they bear in the neighborhood of the pitch point than elsewhere. Here the action is practically one of pure rolling without the sliding of one surface over the other, so that the assumption made in working out the mathematical theory is almost strictly true for the region in which the surface of the teeth is most heavily loaded, and in this region the distribution of pressure represented by Fig. 5 is possible.

In that case the load carried by a tooth is:

$$W = \frac{4.896\,\lambda\, u_0\, R}{h_0} \text{ dynes per cm. run}$$

If R and h_0 be expressed in inches and u_0 in feet per second, this becomes:

$$W = \frac{\lambda\, u_0\, R}{1{,}180\, h_0} \text{ pound per inch run} \quad . \quad (1)$$

Here, as before, λ is to be taken in absolute C.G.S. units.

The maximum pressure occurs in the oil film at a point where $\frac{x}{\sqrt{2\,R\,h_0}} = 0.47517$, and is given by the relation (in C.G.S. units):

$$P_{max} = 1.521 \cdot \frac{\lambda\, u_0}{h_0^2} \cdot \sqrt{2\,R h_0} \text{ dynes per sq. cm.}$$

or, taking u_0 in feet and linear dimensions in inches:

$$P_{max} = \frac{\lambda\, u_0}{3{,}780} \cdot \frac{\sqrt{2\,R\,h_0}}{h_0^2} \text{ pound per square inch} \quad . \quad (2)$$

Taking a case in which $\lambda = 0.7$, $u_0 = 20.3$ feet per second and $R = 2.75$ inches we find from (1) that if the load on a tooth amounts to 267 pounds per inch run, then h_0, the least thickness of the oil film, is $\frac{1}{8{,}050}$ inch and with this value of h_0 equation (2) gives 6,376 pounds as the maximum pressure within the oil film.

From (2) the stress on the metal varies proportionately to the square root of R, provided that there be a definite minimum limit to the thickness of the oil film. Observations made during the past four years prove, however, that lubrication may be perfectly maintained under oil film pressures so high that the metal flows. Thus in motor-car worm gears, although the efficiency of some of these gear is 96 per cent or so, the phosphor-bronze of the wheel gradually flows under the pressure in the oil film, which must therefore attain a limit of full 30 tons per square inch. Moreover, the late Lord Rayleigh proved that a film only a few molecules thick could act as a lubricant, and since a film of oil 100 molecules thick would measure less than one-millionth of an inch, it is obvious that in general there is no practical limit to the minimum thickness of an oil film, as failure from too high a stress on the metal will occur long before the film becomes too thin for Reynolds' equations to be applicable.

Recent experiments have indicated that the difference between a good lubricant and a bad one lies in a good lubricant adhering more firmly to metallic surfaces than does a bad one. A good lubricant welds itself to the surface lubricated, and it seems probable therefore that the lubrication breaks down when the shearing stress on the film is so great as to break the bond so that the ultimate layer of the lubricant slips over the

metal surface, whilst the theory assumes that there is no slip. If this view be well based the permissible load on gear teeth per inch run should be proportional to R as above defined, instead of in proportion to the square root of R.

The shear is a maximum at the point of inflection of the pressure curve, which for the case represented in Fig. 5 occurs on the "leading" side of the point of contact, when

$$\frac{x}{\sqrt{2Rh_0}} = 1 \cdot 205$$

whilst the point at which the opposing surfaces most nearly approach each other occurs when $x = 0$.

For the case already taken above, in which $R = 2 \cdot 75$ inch, $h_0 = \frac{1}{8,050}$ inch and $u_0 = 20 \cdot 3$ feet per second, and the load carried is 267 pounds per inch of tooth, the distance of the point of maximum shear from the point of minimum thickness is 0.032 inch. This value affords some idea as to the effective width of the oil film, since as Fig. 5 shows the pressure falls off rapidly beyond the point of contrary flexure.

The value of the maximum shear is:

$$S_{max} = 1.225 \frac{\lambda u_0}{h_0} \text{ dynes per sq. cm.}$$

or, taking u_0 in feet per second and h_0 in inches, we have

$$S_{max} = \frac{\lambda}{4,696} \frac{u_0}{h_0} \text{ pound per square inch} \qquad (3)$$

For the case already cited this equation shows that even under the very moderate loading and speed there assumed the shear stress tending to make the lubricant slip over the metal amounts to about 24½ pounds per square inch. By substituting from equation (1), equation (3) can be put in the form:

$$S_{max} = \frac{0 \cdot 251 \, W}{R} \text{ pound per square inch}$$

so that if the limit of endurance is fixed by this maximum shear W should vary directly as R.

If hammering is unavoidable the damage done may well increase faster than the load on the teeth per inch run. For a given elastic system the energy stored in it varies as the square of the load. It is this energy that causes the teeth to separate, and it is at least a plausible assumption that the damage done on the return of the teeth should be proportionate to the energy expended in separating them. The moral would seem to be that cyclical hammering of the teeth is to be avoided at all costs, and that if this is done the load imposed may increase directly as the equivalent radius of curvature of the teeth surfaces.

The rise of temperature as the oil passes between the teeth of the gear is proportional to:

$$\frac{W^{\frac{3}{4}}}{R \cdot \sqrt{u_0 \lambda}}$$

and this may affect the adhesion of the lubricant to the metal. It may be prudent, therefore, to increase W, the load on the gear teeth, rather less than in direct proportion to the value of R.

In establishing the mathematical theory of lubrication it was assumed that the change of curvature of the surface of the teeth at the point of contact could be neglected. An approximate calculation of the amount of this elastic distortion shows that the assumption in question was perfectly sound.

To sum up, a review of the evidence at present available appears to indicate that recent gear troubles on shipboard have not been attributable to excessive loading of the teeth, but to the use either of too heavy turbine rotors, or to lack of inertia in the final wheel of the transmission. It would seem that the object of the designer should be to reduce to a minimum the angular momentum of the turbine rotor, and to increase the angular momentum of the last wheel of the gear as much as is commercially practicable. Unfortunately, the conclusions at which we have arrived are qualitative only. What is needed are figures setting forth the angular momentum of the turbine rotors and of the propeller shaft and wheel in the case of gears which have succeeded, and similar figures for gears which have failed. A comparison of these might yield information of the utmost importance to those responsible for designing marine reduction gears.—"Engineering."

ON DEAD-WATER.

The peculiar effects of shallow water on the performances of steamships and of destroyers in particular are well known. Occasionally phenomenally good results have been obtained on trials over shallow courses and at other times unexpectedly bad results were given, depending on the depth of water, the size of vessel and the speed. These peculiarities in performance called for investigation and numerous trials have been made by vessels over deep and shallow courses, and many experiments have been carried out in experimental tanks with varying depths of water. It was found that certain depths had to be allowed if normal performances were to be expected; accordingly, certain courses have been discarded and others approved of.

A still more peculiar phenomena has been observed in certain waters, the cause of which for long was a course of speculation. This is known as dead-water and the experiences of Nansen in the *Fram* in his polar expedition brought about an inquiry which resulted in the mystery being solved.

Nansen's experience was as follows: The *Fram* was in open water and steaming in calm weather. Thick ice was ahead which blocked further progress. The ice was approached to make fast to it, but the *Fram* got into dead-water and made hardly any way in spite of the engine going at full pressure. Her common speed in smooth water and at full steam pressure was at that time fully 4½ knots. This was reduced to about 1 knot when in dead-water. Nansen commented thus:—"We could hardly get on at all for the dead-water, and we swept the whole sea along with us. It is a peculiar phenomenon. We had at that time a better opportunity of studying it than we desired. It occurs where a surface layer of fresh water rests upon the salt water of the sea, and this fresh-water is carried along with the ship, gliding on the heavier sea beneath, as if on a fixed foundation. Dead-water manifests itself in the form of larger or smaller ripples of waves stretching across the wake, the one behind the other arising sometimes as far forward as almost amidships.

We made loops in our course, turned sometimes right round, tried all sorts of antics to get clear of it, but to very little purpose. The moment the engine stopped it seemed as if the ship were sucked back. * * * In two watches (8 hours) we hardly advanced 8 miles, the engine constantly working at full speed."

Whilst the bulk of the evidence relating to experiences with dead-water comes from Norway, where the presence of layers of fresh water on top of the sea water can be reasonably expected, evidence of peculiar phenomena taking place has been received from officers of vessels trading on the estuaries of rivers in America, Africa, Europe and Asia.

The problem was put before Prof. Bjerknes by Nansen and he in turn suggested to one of his pupils, Dr. Walfrid Ekman, that he should undertake experiments and try to account for the phenomena. It was anticipated by Ekman that waves in the salt-water fresh-water boundary should cause effects quite similar to those recorded of dead-water. It was not thought that such should prove so powerful in effect as was actually discovered.

The experiments were carried out to imitate, as closely as possible on a small scale, real cases. A model of the *Fram* and some other boats were towed in a glass tank filled with homogeneous water or having water-layers of different specific gravities. The towing-force and the velocity of the models were registered, and one of the water-layers being colored, the wave motion at their common boundary could be observed and photographed. The experiments thoroughly confirmed Prof. Bjerknes' opinion: the vessel when moving at low speeds generated large waves at the salt-water fresh-water boundary and the resistance at these speeds was enormously increased. At higher speeds, however, the waves disappeared and the resistance was not affected by the fresh-water layer.

The term dead-water was applied by Norwegian seamen to this phenomenon, but the same term has been used in another connection by seafaring men. In a paper by Mr. W. Froude on "Apparent Negative Slip in Screw Propellers" (Transactions of the Institute of Naval Architects, vol. viii), the operation of the dead-water on the propeller is given as a possible explanation of apparent negative slip. Froude said: "What is meant by dead-water is the volume or volumes of water collected behind the sternpost of a ship when in motion—and behind the square tuck when she has one—water which accompanies the ship almost without change. When the sternpost is thick and the tuck broad, as in our old timber line-of-battle ships, the volume of water thus carried is very large; indeed, it was mentioned to me by one of the wardroom officers of such a ship, that when the screw was hoisted out of the water and the ship going 7 knots under canvas, he had been lowered down into the well and had bathed there without inconvenience."

The difference between this dead-water and wake is that whereas the wake consists of constantly renewed volumes, the dead-water having been once for all put in motion accompanies the ship almost without change. In modern ships this kind of dead-water is non-existent.

Although the peculiar effects on power and speed of a layer of fresh water on top of salt are seldom met with in our waters, the phenomenon enters into the navigation of the Norwegian fjords and rivers to some extent. Some such phenomena have also been observed in the Panama Canal where mixtures of waters of different densities have created currents and caused alarm and astonishment at the behavior of ships navigating it.—"Shipbuilding and Shipping Record."

LUBRICATING OIL SYSTEM FOR GEARED TURBINES.

Considering the vast amount of difference between the equipment and operation of a ship fitted with reciprocating engines and one fitted with geared turbines, it may be regarded as indicative of the thoroughness with which the work of building and running vessels fitted with the latter type of machinery has been carried out, that so few breakdowns have occurred. In the United States, as is well known, the operating troubles with geared turbines have been recorded to an extent which was considered in official circles as being abnormal. Many of these breakdowns were attributed to faulty operation or failure of the lubricating system, and in consequence it was decided by the Emergency Fleet Corporation to study all the oiling systems in use on American ships with a view to developing a standard system for use on all geared turbine vessels building or to be built for the Emergency Fleet Corporation. While troubles have not been met with to such an extent with geared turbine vessels built in this country, it is widely recognized that the lubricating system plays a vital part in the successful operation of machinery of this type, particularly where double reduction gears are employed, and a study of the distinctive features of the system adopted by the Emergency Fleet Corporation may, therefore, prove of value.

Particulars of the standard lubricating oil system for geared turbines adopted by the Emergency Fleet Corporation were recently given in a paper read before the American Society of Naval Architects and Marine Engineers, from which it appears that the system in its final form was not issued until July 15, 1919, that is to say, considerably after the war was over and when any limitations or emergency equipment imposed by the probable effects of hostilities could be eliminated. As finally approved the system is of the gravity-feed type, having a working pressure in the region of 12 pounds per square inch. It is therefore specified that the gravity tanks shall be located in the engine-room casing, so that the bottoms of the tanks shall be 30 feet above the center line of the turbine. This, it is stated, has proved very satisfactory in service and no trouble has been experienced in securing the required pressure. There are two of these tanks, each of 600-gallon capacity, for a vessel of 3,000 horsepower, each being provided with heating coils for heating the oil to a temperature of 180 degrees Fahr. preparatory to filtering. The oil is discharged from either of these tanks through an internal pipe extending 8 inches above the bottom of the gravity tank to a line leading to all the bearings and gears, and from thence it is drained to the sump tank. In addition to the main supply pipe, a 1½-inch drain pipe is led from the lowest point of the gravity and reserve tanks through an oil filter or separator to the drain tank, so that oil may be filtered continuously or at stated periods. An overflow pipe is also run from the side of the gravity tank and as near the top as possible to the drain tank, so as to give the air a good chance to be released from the oil through the vent pipes. Further, a connection is provided on the oil discharge line to enable the oil to be discharged direct to the bilge in the event of the oil becoming badly emulsified. The piping arrangements permit either gravity tank to be used as a settling tank.

On the question of filtering the oil the specification is not absolutely rigid; it is essential that some type of strainer, filter or separator shall be fitted, located on the discharge line adjacent to the pumps. If the ordinary type of duplex filter is fitted, this must be placed in such a position as to permit the removal of the strainer baskets by lifting them upward, care being taken that the strainers are located in such a manner as to prevent the sediment falling back into the strainer on removal of the

baskets. They must, of course, be so designed that one side can be cleaned while the other is in operation. It is recommended, however, that a centrifugal type of separator should be installed in lieu of the usual filter or separator, since the results obtained with the centrifugal separator show greater efficiency as the oil, water and sediment are separated, the oil flowing from one nozzle, water from another and the overflow mixture from another, while the sediment remains in the bowl of the separator from which it can be removed at will. Tests have proved that the rolling or pitching of the vessel has no ill effects on the successful operation of the centrifugal separator, and it has been found particularly effective in breaking up an emulsion of oil and water.

Prior to the drawing up of the standard specification it was found that nearly all the oil coolers used were inadequate to the duty they had to perform. This was partly due to the fact that the manufacturers of reduction gears had overrated the efficiency of the gears. It was considered essential, therefore, to fit an auxiliary cooler so that not only could the oil be cooled by this cooler in the event of the main cooler being rendered inoperative by leaky tubes or other causes, but the auxiliary cooler could also be used in parallel with the main cooler when the vessel is passing through tropical waters, where a cooling water temperature of 84 degrees Fahr. or even higher is obtained. The two oil coolers specified are of the same size and capable of cooling 175 gallons of oil per minute through 20 degrees Fahr. They are equipped with piping to enable the coolers to be by-passed, the oil being discharged directly from the pump to the gravity tank, this precaution having been found necessary when using heavy oils with cooling water of low temperature.

The question of safety devices is one of great importance, and in addition to the usual gages, glasses, thermometers, flow indicators, &c., it was considered advisable to fit an automatic protective device. This took the form of an electric alarm system actuated by a float switch in the gravity tanks. This notified to the engineer on watch and to the chief engineer in his room that the oil had dropped below the normal operating level. In addition, it also automatically increased the speed of the pumps. It was deemed advisable to omit this in later vessels owing to the necessity of reducing the cost of the installation rather than because of any belief in the lack of necessity for it, and in fact it is recognized that in almost every respect the standard specification errs on the side of liberality and high cost. It is stated, however, that if an increase of efficiency is obtained with reduction gearing in the future, as it should be, this will permit of a considerable reduction in the size and consequently the cost of the lubricating system.—"Shipbuilding and Shipping Record."

U. S. NAVAL AIRCRAFT.*

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In the few years that Naval Aircraft have been part of the Navy they have become so specialized by adaptation to meet specific naval requirements that they constitute today distinctive types of design. Before considering these peculiarly naval types, let us examine the naval functions which have governed their development.

* Presented at the joint meeting of the Franklin Institute and the Aero Club of Pennsylvania held Wednesday, January 21, 1920.

The Navy consists of two major parts: the Fleet, which is designed to operate on the high seas in any part of the world, and the Shore Establishment, organized as Naval Districts on the coasts of the United States and the outlying possessions. The primary function of the Fleet is to discover, engage and destroy the enemy fleet. The functions of the Shore Establishment are both industrial and military. The industrial function does not concern us here. The military function is the elimination from coastal waters of the activities of enemy naval forces, in particular, raiders, submarines and mine-layers.

The Navy employs every weapon or device which is considered to be useful in connection with its mission. As the mechanical arts advance, the latest applications are introduced. Progress in any of the arts or sciences is eagerly seized upon to discover means for making use of the new knowledge in the Navy. So with aeronautics; as the art has progressed, aircraft have been developed for use in the Navy. The principles governing the design of such Naval Aircraft are the very fundamentals of the art, but the aircraft themselves represent the application of these principles to meet the needs of the Navy; that is, to fulfill functions allied to the functions of the Fleet and the Shore Establishment.

Consider the Fleet first. The Fleet must have scouts to discover the enemy and give intelligence of his strength and probable intentions. Aircraft may obviously be used for this purpose in conjunction with scout cruisers, destroyers, or battle cruisers. The range of visibility from aircraft is relatively enormous, as an observer in an airplane can go three times as fast and see three times as far as an observer on a ship. With this inherent advantage, it only remains to develop practical scouting or reconnaissance aircraft for such work. This development results from no idle arm-chair speculation, however, as the problem is limited by the restriction that the Fleet is normally at sea and the weather normally bad.

Scouting, or reconnaissance for the Fleet, may be of two kinds: First, reconnaissance at a great distance, requiring great endurance in the air; and second, reconnaissance at a short distance immediately in advance of the Fleet. For this second function, great endurance is not needed as a frequent return to the parent ship is possible. We thus will have at least two types, one characterized by extreme endurance, the other by simplicity and handiness for use from shipboard.

Another distinct type must be employed for fighting enemy aircraft which, in any future naval engagement, will surely be present. This problem presents considerable difficulty as it presupposes a battle in the sky for control of the air in advance of the actual engagement of the ships. Control of the air over a wide area can only be maintained by large numbers of airplanes of the greatest fighting capacity and the launching of a swarm of airplanes at the desired moment from the restricted spaces aboard ship seems impracticable. Consequently, special ships in the Fleet with flush decks and enormous hold spaces are necessary to act as carriers.

Before the introduction of aircraft into the Fleet, the control of gunfire was directed by a "spotter" stationed on the mast. This is the highest point on the ship from which to observe the splash of the shells, but compared with a position several thousand feet in the air, it is very low—too low, for example, to see over a smoke screen made by enemy destroyers, and too low to estimate at extreme ranges the distance from the target to the splash. The individual battleships of the Fleet have need, therefore, for aircraft to carry the fire control officer or spotter aloft.

The aircraft so far discussed are primarily concerned with air fighting and observation. There remains the offensive function which is the

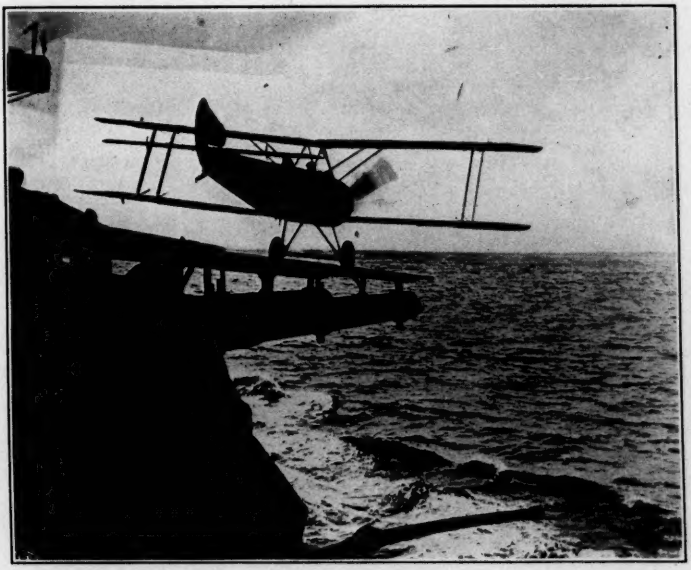

FIG. 1.—SHIP'S FIGHTING PLANE FLYING OFF TURRET PLATFORM.
(U. S. NAVY.)

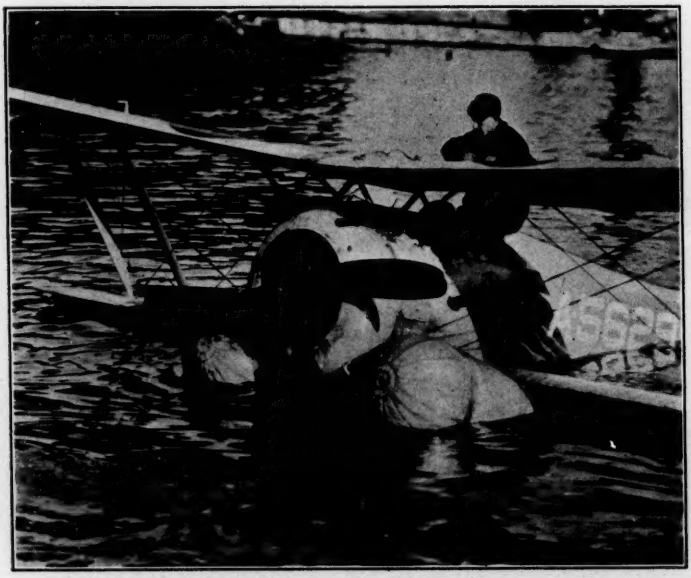

FIG. 2.—SHIP'S FIGHTING PLANE AFTER A WATER "LANDING." NOTE INFLATED AIR BAGS. (U. S. NAVY.)

FIG. 3.—H. M. S. *Argus*, AIRPLANE CARRIER. (BRITISH NAVY.)

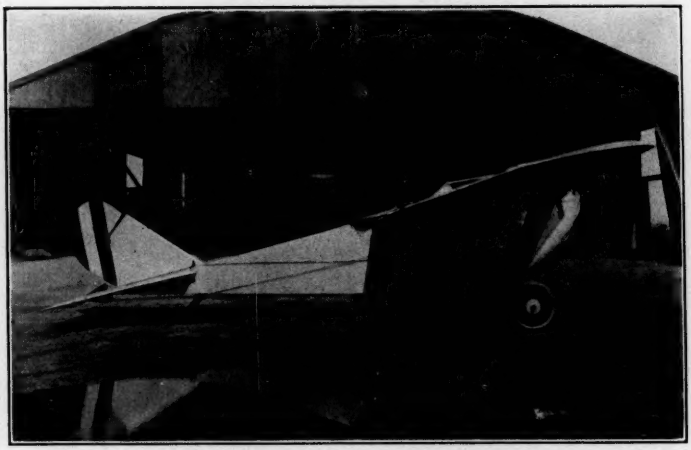

FIG. 4.—SHIP'S SPOTTING PLANE. (U. S. NAVY.)

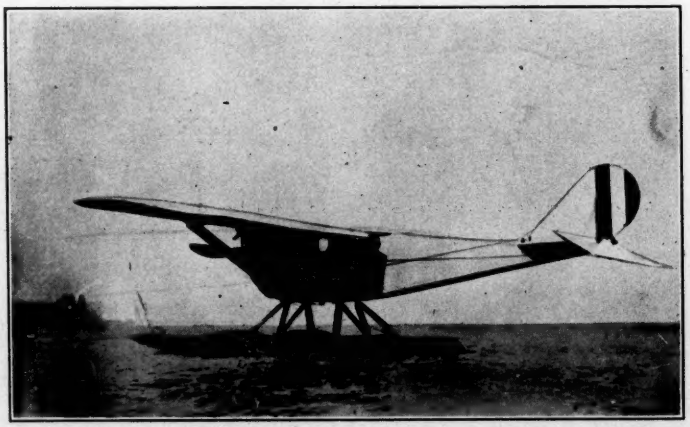

FIG. 5.—SHIP'S SPOTTING PLANE. SAME MACHINE AS FIG. 4, FITTED WITH FLOATS. (U. S. NAVY.)

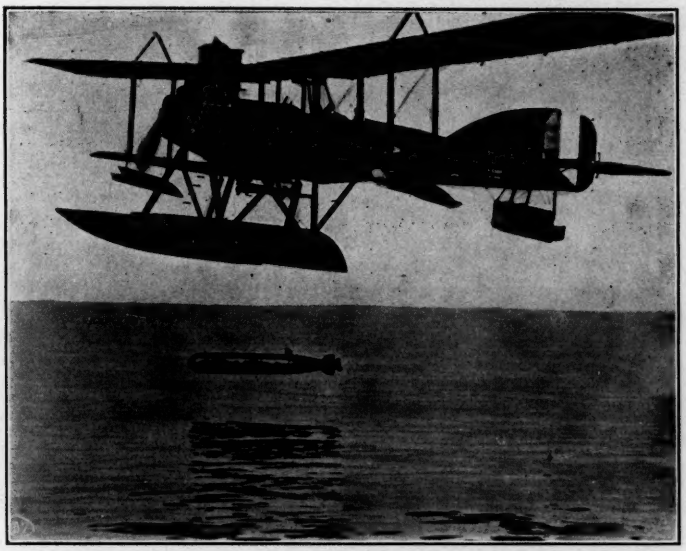

FIG. 6.—TORPEDO PLANE AT INSTANT OF LAUNCHING TORPEDO. (BRITISH NAVY.)

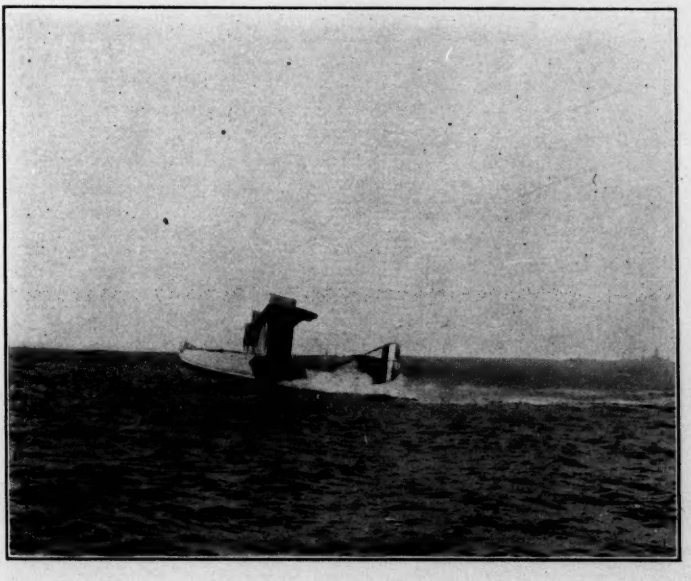

FIG. 7.—COAST PATROL AND CONVOY TYPE. FLYING BOAT F-5 AT START. (U. S. NAVY.)

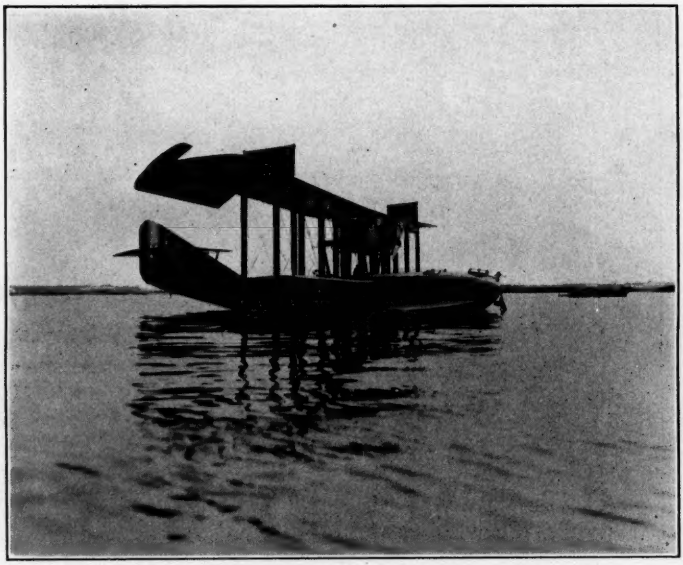

FIG. 8.—COAST PATROL AND CONVOY TYPE. FLYING BOAT F-5 AT ANCHOR. (U. S. NAVY.)

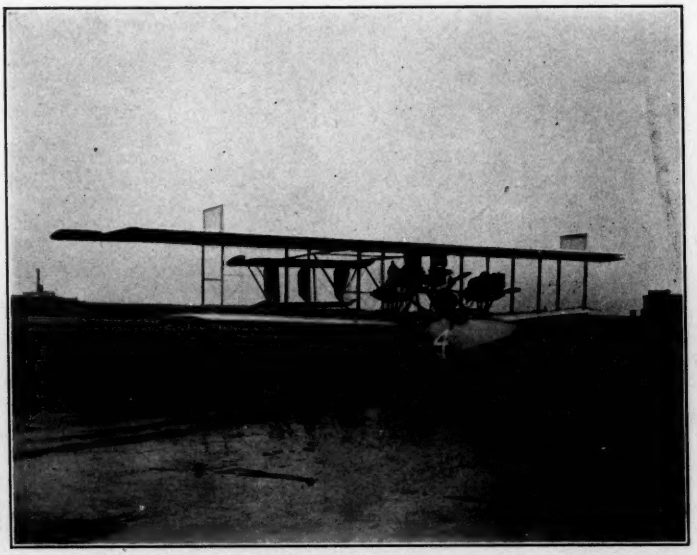

FIG. 9.—FLYING BOAT NC-4, LONG-RANGE SCOUTING TYPE. (U. S. NAVY.)

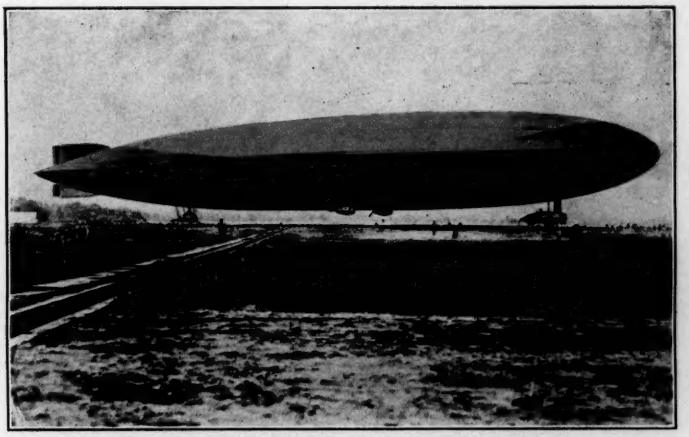

FIG. 10.—ZEPPELIN L-64. (GERMAN NAVY.)

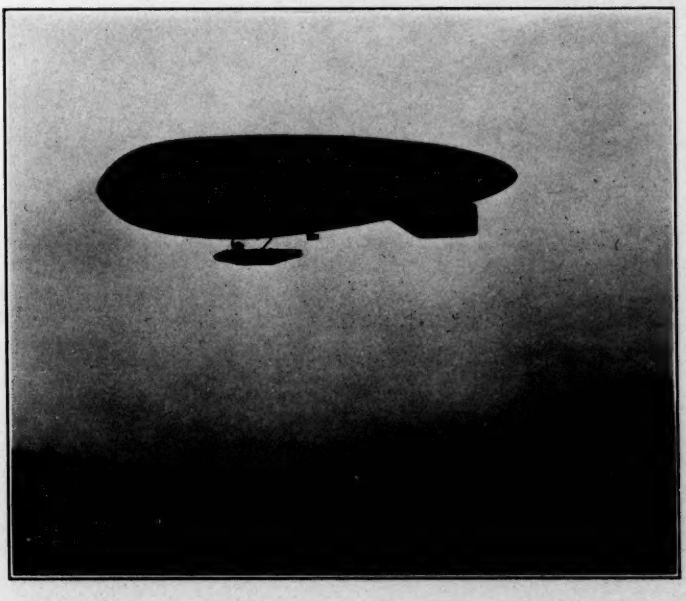

FIG. 11.—NON-RIGID AIRSHIP C-1. (U. S. NAVY.)

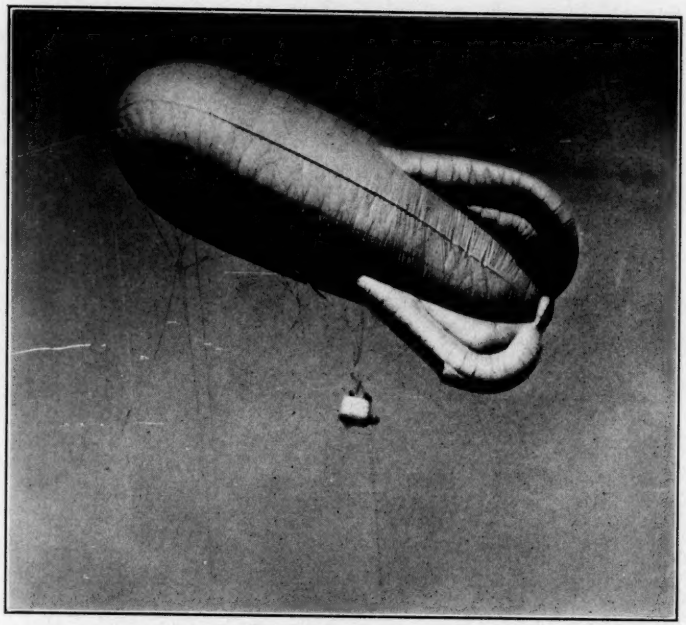

FIG. 12.—KITE BALLOON. (U. S. NAVY.)

primary objective—to destroy the enemy fleet. Aircraft can now carry bombs powerful enough to damage a battleship or a regulation torpedo sufficient to sink her. Torpedo attack from the air cannot be stopped by the usually effective screen of destroyers and light cruisers. In fog or at night the great speed of torpedo-carrying airplanes adds so much to the element of surprise that the chances greatly favor the attack. Furthermore, booms, nets and chains across a harbor mouth will no longer render the anchorage immune from attack. The torpedo plane during the War was not developed to full effectiveness by either side, but in any future war we may expect it to play an important part.

To recapitulate, there are five essentially distinct functions for aircraft with the Fleet:

1. Long-range scouting.
2. Short-range scouting.
3. Fighting in the air.
4. Spotting or control of gunfire.
5. Bomb and torpedo attack.

The Shore Establishment divided into Naval Districts has, likewise, uses for aircraft in connection with the control of coastal waters. The first and most important is searching for hostile raiders, submarines, mine-layers, etc., which duty may be termed "Coast Patrol."

In connection with coast patrol it may become necessary to convoy shipping into and out of ports and through dangerous waters. For this purpose aircraft are essential, not alone of course, but co-operating with an escorting force of destroyers.

In addition to the applications of aircraft mentioned so far, we have very necessary aircraft work required in connection with Advanced Base and Expeditionary Forces. Such Expeditionary Forces may usually be supposed to operate without opposition from a highly developed air force, but such a favorable condition cannot be guaranteed. The Marines, therefore, have to be provided with the latest thing in military airplanes. Such types are similar in function and hence in design to those used by an army, and I shall not take time to discuss them as Naval Aircraft.

In the foregoing, I have briefly classified aircraft functions as based on the functions of the Navy. In considering how aircraft can fulfill these functions we are immediately confronted with the necessity for deciding whether we require "heavier-than-air" or "lighter-than-air" types.

Under "heavier-than-air" we include all airplanes and seaplanes which are supported by their own wings. By "lighter-than-air" we designate balloons and airships or dirigibles. Kites and parachutes are not classified as aircraft.

Among "heavier-than-air" craft, we have: First, airplanes with wheels of the ordinary type; second, seaplanes with floats or pontoons to fit them to start from and alight on the water; and third, flying boats, being a true boat fitted with wings. These types may be somewhat mixed in special designs as, for example, a seaplane provided with wheels as well as floats to make it amphibious, or an airplane fitted with hydro-vanes and air-bags to permit emergency alighting on the water.

"Lighter-than-air" craft consist of the ordinary free balloons, or "sphericals," kite balloons of elongated form which are towed by a cable, non-rigid airships (or dirigibles) obtaining lift from a gas-bag under pressure, and rigid airships such as the Zeppelin with a rigid hull structure fitted with a multiplicity of gas-bags.

In the present state of the art we may choose one type or the other for a certain naval function, depending on clearly evident practical con-

siderations, but for others, it is not clear whether we should favor an airship or a large seaplane. The final decision can only be made after experience with each, but by that time progress in the art may require a reversal of the conclusions reached. It is wise, therefore, to provide both and allow the stimulation of competition to develop the possibilities of each.

Let us now look at the types of aircraft already in use, and consider what is the state of the art, the requirements of the service, and the possibilities for meeting them in the immediate future.

For long-distance scouting at sea, it is primarily a question of the greatest endurance and radius of action. There is no aircraft which approaches the rigid airship in these characteristics. The present-day German and British rigids of 2,000,000 cubic feet volume have a maximum radius of action of over 3,000 miles and can stay out about five days. The best that has been done to date with a flying boat is about 1,500 miles. While the radius seems to be half that of the airship, the comparison is less favorable when it is considered that the flying boat cannot keep in the air quite a day. On the other hand the airship can drift about while waiting or observing without consumption of fuel; the crew are comfortable and probably will do better work; radio is more powerful and communication with the Fleet more certain.

The rigid airship, according to Admiral Jellicoe, is worth two light cruisers for scouting in favorable weather. The last words must be pondered, for "favorable weather" is a relative term. Fog is no doubt most unfavorable weather for a seaman, but for an airship at sea low fog may give it a distinct advantage in scouting. The airship will, however, have to cope with storms and heavy rain and snow. Due to her great endurance and the usually local character of really violent meteorological disturbances, the airship can be expected to avoid getting into trouble. Naturally, we must assume adequate weather reports by radio from the ships co-operating with her. However, the necessity to avoid storm centers will put a stop to any scouting for the time being. A flying boat is relatively unaffected by storms while in the air, but she cannot alight on the water under such conditions. With present power plants such a possibility must always be kept in view.

The latest German rigid airships have a maximum speed of about 70 land miles per hour and the NC-4 flying boat has a top speed of about 90 miles per hour. To cope with gales of general extent the airship is clearly at a disadvantage. It is probable that, following the present trend of design, the airship speed will not be pushed much, if any, above 80 miles, while a speed of 115 miles for the flying boat is already in view. It seems evident that the relative utility of the airship and flying boat depends upon weather conditions. For scouting over the North Atlantic in seasons of high winds, the flying boat may be the more suitable.

I have not here considered large land type airplanes which, while comparable with large flying boats in speed and endurance in the air, are unfit to send out to sea, as they cannot land to refuel or make repairs, but must make non-stop flights from land to land. The scouting requirements of the Fleet are unlikely to be always in waters conveniently near the aerodrome. The flying boat will have to be based on shore for hauling out and overhaul, but normally she will follow the fleet, being supplied with fuel from a supply ship, except in rough weather, when she will take refuge in the nearest sheltered water like any small craft. There is no need for an expensive and elaborate chain of landing fields, as all the bays and rivers of the world are "landing" spots more level than the most perfect field on shore.

The rigid airship, like the airplane, must start from and return to a land base, but her five-day endurance minimizes this restriction. There must be, however, airship bases located at all strategic points near which the Fleet is likely to operate.

The cost is interesting for comparison. The rigid airship of the greatest size yet constructed costs about twenty times as much as the NC-4, the largest flying boat yet built. The shed to house her also costs about twenty times as much as housing on shore for NC-4. Considering cost alone, we should hope the airship were as useful as twenty flying boats. Under some conditions this may be the case. Any number of flying boats bringing in negative reports is worthless compared with continuous positive and definite information from an airship keeping the enemy under constant surveillance.

It must not be forgotten that the rigid airship of the present day is very vulnerable to attack from airplanes, as has been demonstrated over London to the entire satisfaction of the Germans. The German bombing raids over London were entirely unjustifiable, either on grounds of ordinary decency or from tactical considerations. The legitimate function of the Zeppelin was scouting over the North Sea where most valuable work was done. Indeed, on more than one occasion, the Zeppelin scouts permitted the German High Seas Fleet to avoid action and certain defeat by the British Grand Fleet.

It is hoped that at some future day helium may become cheap enough to be used in place of hydrogen. Until that time comes, rigid airships must avoid coming within the range of enemy ships or aircraft. At sea, with its immense distances this is quite possible to accomplish. There is no need to come within range of a ship's guns in order to observe its movements and, in general, airplanes can be avoided by rising into the clouds. The airship, at high altitudes, can rise faster than an airplane as the latter climbs by engine power, which is becoming weak, while the former climbs by dumping ballast, which is always effective. Furthermore, at high altitudes, the airship suffers no loss in speed while the airplane, no matter how fast at sea level, loses speed rapidly with altitude. During the chase of an airship it is also to be remembered that the airplane may come under the fire of several guns of the airship. On the whole, the airship should be able to keep clear of hostile aircraft.

The present state of the art indicates the rigid airship and the large flying boat as rivals as scouts for the Fleet. The rigid airship in its latest development is represented by the Zeppelin and the flying boat by the NC-4 type of our own Navy. Is it likely that the possibilities of the immediate future indicate any important change in the relative merits of the two types?

Curiously enough, it has been proposed to build a rigid airship twice the size of the present-day Zeppelins and at the same time the flying boat adherents have proposed to double the NC-4. Such an increase in size at one step appears to be the maximum it is safe to attempt and it will probably take the next few years to see light ahead. From a preliminary analysis, however, it appears that increase in size is more favorable to the airship than to the flying boat, and as we build larger and larger airships the flying boat is left farther behind.

I shall not burden this paper with the detailed analysis, but fundamentally the difference results from the fact that for the airship the buoyancy of the hull is increased as the cube of a linear dimension, while for flying boat wings the lift remains the same per square foot or increases as the square only. As airships are made larger the margin of the total lift available to carry fuel and stores is increasing, and an endurance of

seven or eight thousand miles is not difficult to reach with an airship of about 4,000,000 cubic feet. In other words, by doubling the size of the airship the endurance may be doubled.

On the other hand, the flying boat, if doubled in size, will be likely to add but 25 or 50 per cent to its endurance. The large flying boat has a decreasing margin of lift available for fuel and stores and an increase in size is uneconomical unless the greater size permits use of better structural arrangements, or better structural material to such an extent as to overcome the unfavorable factors which are inherent in the problem. The effort of the designer will be to provide the larger flying boat with no worse a ratio between useful load and total weight than we have today on the NC-4.

The principal advantage of the large airship as compared with the large flying boat is derived from more economical propulsion. The resistance to propulsion of an airship is largely due to the hull or envelope and it is known from wind tunnel experiment and confirmed by experience that this resistance is represented by an expression of the form:

$$R = KS^{\frac{n}{2}} V^n$$

where n lies between 1.86 and 2, and near 1.9. To simplify the argument, let us take $n = 2$, on the safe side. Then $R = KSV^2$ where S is the "wetted surface" or surface exposed to air friction, V the velocity and K a coefficient depending on form. The resistance thus varies as the square of a linear dimension and the square of the speed. The necessary power of the engine is:

$$P = KL^2V^3$$

But the lift depends on the total volume which varies as L^3 and hence the tons of weight per horsepower vary as:

$$\frac{\text{Lift}}{\text{Power}} = \frac{L^3}{KL^2V^3} = \frac{L}{KV^3}$$

For larger ships this ratio becomes more favorable. Since the proportion of the total lift available to carry fuel is known to be more favorable in such large ships as have been built, it is on the safe side to consider such proportion constant for purposes of argument. The weight of fuel then varies as the total lift or as L^3. The endurance in hours is the quotient of fuel available and fuel consumed per hour. This quotient is proportional to the ratio given above, or to tons per horsepower:

$$\frac{L}{KV^3}$$

The larger ship will go farther at the same speed in a manner at least directly proportional to the length.

As a matter of fact, experience has shown that the larger ships actually have a greater fraction of the total lift available for fuel. Nor is the resistance so high as assumed. Therefore, the increased endurance of larger airships is even greater than would be predicted from this simple analysis.

The following table of German Zeppelins, illustrates a progressive improvement with size, but it cannot be expected that such improvement will be continued indefinitely at the same rate, for we should eventually reach a ship such that the empty weight was practically nothing. The rather abnormal improvement shown is due to the fact that the early ships were

the small ones and their design was rather crude and heavy. The larger ships are late designs of great refinement. However, the improvement is significant:

Type	Date	Volume cubic feet, 100 per cent. full	Length, feet	Diameter, feet	Lift		Useful				Endurance	
					Gross tons	Useful tons	Gross, per cent.	B. H. P., total	Speed, full power, M. P. H.	Full power, miles	Half power, miles
L-1	1912	793,000	517	48.5	22.8	....	....	540	47	....	....
L-2	1913	918,000	525	54.0	26.4	7.3	27.6	800	49	1100	1750
L-3-9 inc.	1915-16	919,000	525	54.0	26.4	....	....	800	49	....	....
L-31-38 inc.	1916-17	1,945,000	643	78.7	56	26	46.5	1449	60	3300	5050
L-30, 40, 43, 45 and 47	1917-18	1,945,000	643	78.7	56	28.6	51	1440	60	3300	5200
L-44, L-46	1917	1,945,000	643	78.7	56	29.5	53	1300	62	4200	6700
L-48-52 inc.	1917-18	1,945,000	643	78.7	56	30	54	1300	62	4652	7350
L-53-64 inc.	1917-18	1,974,000	643	78.7	57	29	51	1500	68	3950	6300
L-70	1918	2,200,000	693	78.7	63.4	35.7	56	2100	80	3900	6300
L-71	1918	2,420,000	743.1	78.7	70	42	60	1800	74	4800	7600
Bodensee	1919	700,300	393.7	61.3	20.3	7.3	36	1040	80.7	1350	2500
R-38 (British)	1921	2,724,000	694.4	86	78.6	45.9	58.4	1950	70	5100	8100

The flying boat cannot be studied in exactly the same manner as the total lift is given, not by the volume of gas, but by the wings. The lift of an aeroplane wing is represented by an expression of the form:

$$\text{Lift} = KSV^2 f(i)$$

where S is the area, V the velocity, K a coefficient depending on the form of the wing and $f(i)$ a function of the angle of attack, i of the wind on the lower side of the wing. It is an experimental fact, so far without adequate explanation, that an aeroplane wing is useful only between angles of 0 and 14 degrees about. Near 14 degrees, there is a change in the nature of the flow over the wing and violent turbulence is set up. For an angle greater than this critical angle the lift is no greater, and may even be less, while the resistance or drag becomes relatively enormous. Consequently, the aeroplane gets its practical maximum of lift at about 14 degrees, and for a given weight to be carried can fly slowest at this attitude. Since the plane must get up to this speed by a run on the ground or over the surface of the water, before she will take the air, this limiting speed is the speed of starting and must be kept within practical limits.

For airplanes of the land type the starting speed is between 45 and 55 miles per hour. If anyone has ever attempted to drive an automobile over a pasture or meadow at such speed, he will have an idea of the risk and danger of getting into the air with a large airplane.

The conditions for a seaplane are somewhat more favorable because the water is more nearly level than the average pasture. With modern boat hulls (like the NC-4) a get-away speed of over 60 miles per hour is quite permissible even in rough water. The seaplane or flying boat can, therefore, carry into the air a greater weight for a given wing area simply because the speed can be permitted to go up.

In general, however, the starting and landing speeds of all heavier-than-air craft are restricted by practical considerations and for this reason we may assume that the minimum velocity in the expression, $\text{Lift} = KSV^2 f(i)$ a maximum, remains substantially constant.

If we build larger flying boats, the ratio $\frac{\text{Lift}}{\text{Surface}}$, therefore, remains constant. The wing area must increase directly as the total weight of the machine.

The weight of larger wings carrying the same unit load of, say 9 or 10 pounds per square foot, naturally tends to increase faster than the area of such wings. The larger wings have a greater spread and are more difficult to brace and keep stiff and strong.

The carrying capacity of a wing increases with the square of a linear dimension, but the structural members of which it is composed vary, some as L^3, some as L^2, and some even as L^4. The simple geometrical expansion of a small wing to make a large one would be bad engineering.

Fortunately, one need not be discouraged by such theoretical considerations as we have learned from past experience that, by the introduction of new and better structural materials and arrangements in larger wings, the weight can be kept down nearly to direct proportionality. That is: the weight of large wings per unit area is the same as for small ones carrying the same unit load.

This result, quite at variance with the theoretical deductions of several learned writers, is illustrated by the following table:

Name of machine	W (Gross weight)	A (Wing area)	$\frac{W}{A}$	Weight of wings per square foot (including struts and wires)
R-6	3,964	613	6.4	1.32
HS-1-L	5,902	653	9.0	1.78
HS-2-L	6,373	803	7.9	1.69
H-16	10,900	1,164	9.36	1.76
Caproni	12,810	1,420	9.02	1.15
F-5-L	13,000	1,397	9.3	1.81
F-3	13,400	1,425	9.4	1.70
F-6-L	13,514	1,397	9.7	1.55
NC-2	22,600	2,441	9.3	1.59
NC-4	28,000	2,441	11.5	1.59
Tarrant	44,862	4,950	9.06	1.93

The variations in the weight of wing structure per square foot of area are no more than would be expected from the different designers of these widely differing machines. A point of interest not brought out in the table is that the structural factor of safety has fallen from 6 in the HS type to 3 in the NC type. This sacrifice is necessary in order to avoid weight. It is also justified in view of the fact that the very large machines are not called on to perform acrobatic movements in the air.

After the wings, the next principal weight group to consider is the hull. The hull of a flying boat is designed to float the machine on the water with sufficient reserve buoyancy to give stability and seaworthiness in a seaway. In addition, the hull must afford hydroplaning surface enough to permit high speed over the water at the "take-off." Experiments in the towing tank, confirmed by practical experience, show that all the requirements for a flying boat hull are met if the volume of the hull is kept

directly proportional to the total weight of the machine. That is to say, the lines of a large hull may be expanded from the lines of a small one with a scale ratio equal to the cube root of the displacement. The structural weights of such hulls will be almost directly as their displacements, or as L^3, where L is any linear dimension. There is, therefore, little relative gain in hull weight in the larger boats.

An analysis of the weights of a number of large land machines indicates that the weight of fuselage and landing gear is about the same as the weight of the equivalent flying boat hull which combines the two functions.

The power plant weight depends of course on the speed required. Very fast fighting planes may have abnormally big engines which will account for an abnormally large proportion of the total weight lifted. Eliminating these, as well as abnormally slow-training machines, we have the following table from which it seems clear that the percentage power plant is independent of size:

Name of machine	W (Gross weight)	$H.P.$ (Total power)	w (Power plant weight)	$\frac{W}{H.P.}$	$\frac{w}{W}$ (per cent.)	$\frac{w}{H.P.}$
R-6	3,964	200	1,159	19.8	29.2	5.77
HS-1-L	5,902	360	1,566	16.4	26.5	4.33
HS-2-L	6,373	360	1,600	17.7	25.1	4.41
H-16	10,900	720	2,831	15.1	26.0	3.92
Caproni	12,810	1,080	4,400	11.8	34.5	4.07
F-5-L	13,000	720	2,876	18.1	22.8	3.98
F-3	13,400	700	2,939	19.1	21.9	4.2
F-6-L	13,514	720	2,893	18.8	21.4	4.01
NC-2	22,600	1,080	5,674	20.9	25.1	5.24
NC-4	28,000	1,600	7,448	17.5	26.6	4.65
Tarrant	44,862	3,000	9,217	15.0	20.6	3.07

Excepting the Caproni, which is obviously over-engined, the table shows that the power plant for general utility machines of long endurance amounts to about 25 per cent of the total weight and the power plant itself weighs about 4 pounds per horsepower. This includes the complete power plant with all accessories, piping, and auxiliaries.

The figure for the ratio, total weight lifted per horsepower, is about 18 for long-distance flying boats, and while it may vary between 15 and 20 in different designs, such variation is independent of size. We may safely conclude from this survey of past experience (as well as from an analysis based on dimensional theory which I shall not attempt here) that the ratio $\frac{W}{P}$ is substantially constant for large flying boats.

Since the weight of the power plant increases as the gross weight of the planes, and we have seen above that the weight of boat hull and wings increase at least as rapidly, the empty weight of the complete plane should also increase as rapdily as the gross weight lifted. The difference between weight empty and gross weight is the useful load available for

crew, fuel, cargo, etc. The useful load should be, therefore, a constant fraction of the gross weight. The following table bears out this statement:

Name of machine	W (Gross weight)	L (Useful load)	$\frac{L}{W}$ (Per cent.)
R-6	3,964	844	22.3
HS-1-L	5,902	1,832	31.1
HS-2-L	6,373	2,113	33.1
H-16	10,900	3,500	32.1
Caproni	12,810	5,110	39.8
F-5-L	13,000	4,750	36.4
F-3	13,400	5,479	40.9
F-6-L	13,514	5,755	42.6
NC-2	22,600	8,900	39.4
NC-4	28,000	12,000	42.8
Tarrant	44,862	18,745	41.8

In this comparison, I have not labored the point as to whether a large seaplane or a large landplane is more advantageous. Without burdening the paper with further tables, I can state that there is essentially no difference between them. The large landplane can be made somewhat lighter structurally than the flying boat, but due to the necessity for providing a slower landing speed the wings must be larger and, hence, heavier. The net result is practically to make the types equivalent in large sizes.

To revert to the comparison with airships, we have the flying boat's power proportional to total weight or to L^3 and, likewise, the useful load or margin available to carry fuel. Consequently, the endurance which is the quotient of these quantities is independent of size and remains constant so far as the effect of size alone is concerned. Practically, the larger flying boats do, however, have a somewhat greater endurance, but this is not due to size but to greater refinement in design. With very large boats it is possible to increase slightly the efficiency of propulsion by using geared-down propellers of large diameter and by more careful reduction of resistance to propulsion. Also the weight of crew and navigational equipment in a large boat is a smaller proportion of the total lift. All economies taken together may give an increase of some 30 per cent in endurance. No further substantial increase can be expected without material improvement in weight and economy of engines. Naturally, such improvements are expected but cannot be counted on today in a design to be built tomorrow. Mere size is not itself going to give great endurance.

On the other hand, we saw how the endurance of the airship increased at least as rapidly as the length. To get greater endurance we need only build larger and, keeping the speed constant, we get automatically relatively enormous increases in endurance.

We can conclude from all this that for Fleet Scouting at sea where long distances are covered, the rigid airship will be the only suitable type of aircraft. Where the distances to be covered are not too great, where the winds are high, and where hostile aircraft are numerous and efficient, the rigid airship yields first choice to the large flying boat of the "Giant" type, to use the word applied by the Germans, to their 1918 machines.

For daylight patrols over coastal waters a large flying boat is more useful than an airship because a greater area can be searched in given time and because less affected by weather. However, it is often cheaper to use a large number of moderate size flying boats for short patrols.

For convoy work, the flying boat is altogether too fast and of too short endurance. To keep in company with a convoy of vessels proceeding at 10 or 12 knots, the airship is most suitable and for this duty a large number of the cheaper non-rigid airships is more economical to employ than the more costly rigids. The latter should be reserved for Fleet Scouting.

The ship's fighting plane must be carried aboard ship and hence is as small and compact as possible. It is mounted on a platform over a turret and when the ship steams into the wind, the plane flies off in a run of some 40 feet. To be sure of getting off, the plane must be light with relatively large wings. The Army fighting planes are not at all suitable for this work, because they require a long run to get into the air. Furthermore, the Navy ship's fighting plane may have to land in the water in case of engine failure. To permit this, the planes are fitted with emergency flotation gear consisting of vanes, to keep the plane from somersaulting, and air-bags inflated from a bottle of compressed air carried in the plane. Normally, the air-bags are tightly rolled up. The air-bags and vanes are provided instead of regular pontoons to save weight. The fighting plane cannot return to her parent ship directly, but must land on the deck of a special flush deckship called a "Carrier."

The ship's spotting plane is similar to the fighter, except that it is a two-seater of less speed and greater endurance. It is to some extent a general utility machine, and provided with guns to fight off hostile aircraft. The spotting plane may be either a land machine with air-bags or a seaplane.

The torpedo and bombing plane is a larger machine equipped to carry heavy bombs or a torpedo. It will be launched from the carrier. The development of the torpedo plane and the tactics involved in its employment, I am not permitted to discuss in detail. It is, however, a very important type and peculiarly Naval.

I have not yet mentioned the Kite Balloon, which while less interesting technically, is of great importance to the Navy. The Kite Balloon is towed from a ship at an elevation of about 1,000 feet. At such elevation, a spotter can control the fire of the ship's guns unless an enemy smoke screen interferes. In that case, the planes must be depended upon. One may ask, why not depend on planes all the time? The answer is: communications. The communications between a plane aloft and the firing ship are by means of radio or some visual signal. At best such signals are difficult to keep clear and when there are many ships firing and many planes aloft, the chances for confusion and interference are very fair. The Kite Balloon observer has a telephone directly to his ship, giving him freedom from interruption and interference.

I have gone into the realm of prophecy a bit in connection with the problem of the rigid airship *vs.* the large flying boat, because this is a real problem now under discussion. A decision cannot be reached from a consideration of the two types as they exist today, as each is certain of further development. So far as such development can be forecast, the advantage appears to lie with the rigid airship.

The other Naval types mentioned involve no such fundamental difficulty as a selection of the type best worth developing. The ship's fighting and spotting planes are adaptations of planes brought out during the War. The medium-sized plane for coast patrol is fairly satisfactory as it exists at present. It was perfected during the war for antisubmarine work.

Similarly, the non-rigid airship for convoy duty is well developed, both in the United States and abroad, and no important changes in type are anticipated.

In general, the line to be followed in the design of Naval Aircraft is fairly clear. As the special types needed to meet Naval requirements become more highly specialized, their difference from the aircraft used by an Army will become more marked. Already, a Naval Aviator has come to be regarded as a Naval Officer first, and an aviator second.—"Journal of the Franklin Institute."

THE GENERAL ELECTRIC TURBO-SUPERCHARGER FOR AIRPLANES.

By Dr. Sanford A. Moss,

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Dr. Moss, who developed the General Electric turbo-supercharger, compares the device with others intended for the same purpose and with supercharging engines. He then explains the design of the General Electric supercharger, and narrates the very interesting history of its development which includes a series of tests on Pike's Peak and also in flight. There is also included a very graphic description of the instruments and method of calculation employed in accurately measuring airplane altitude. The author concludes his article with a description of the performance of the supercharger.—Editor.

INTRODUCTION.

An airplane flying at high altitude is in an atmosphere of comparatively low density. For instance, at 20,000 feet altitude the density is practically half that at sea level. This means that a given volume contains half as much actual air by weight. The cylinders of an airplane engine are therefore charged with an explosive mixture which has about half the value of a charge at sea level. The engine actually delivers about half of its sea level power at 20,000 feet.

Both the low temperature and the decreased pressure at high altitude have effect in fixing the high altitude density. Both the decrease of temperature and the decrease of weight of the charge affect the carburation at high altitude. The fixed clearance volume and the decreased initial pressure give a decrease of compression pressure resulting in a loss of efficiency. There is, therefore, a combination of causes which gives as a net result the decrease in engine power very nearly proportional to the decrease in density.

At high altitude, the resistance of the air to the motion of the airplane is decreased directly in proportion to the decrease of density. The power required for a given airplane speed is therefore greatly reduced. However, the engine power has been so reduced that the usual net result is a considerable decrease in airplane speed. When the engine power is maintained at the sealevel value, there is, however, a considerable increase of speed at high altitude.

Filling the cylinders of an internal combustion engine with a charge greater than that which would normally occur, is called "supercharging." Methods of doing this have engaged the attention of a great many experimenters.

The "gas turbine" is a prime mover in which highly heated products of combustion impinge directly on a turbine wheel. The high thermal efficiency of the gas engine and the rapid displacement of the reciprocating engine by the steam turbine have caused a great deal of effort to be spent upon some combination of the two in the form of a "gas turbine." Many inventors have proposed various types of gas turbines and a number of these have been developed to the point where their operation is successful mechanically. However, no type has yet shown sufficiently good efficiency to warrant commercial use. The engineers of the General Electric Company have very closely followed the various gas turbine developments and have been intimately in touch with the situation for many years.

In 1903 the Company first began work on the "centrifugal compressor." This is an apparatus similar to the fan blower except that the shape of the impeller blades and the passages leading air to and from the impeller are so arranged as to give efficiency very much greater than that of the usual type of fan blower, so that the apparatus forms a satisfactory means for compressing air to appreciable pressures. A line of single-stage centrifugal compressors has been developed for compressing air from one to five pounds per square inch above atmosphere, to be used for many industrial purposes; as well as a line of multi-stage machines for compressing air and gas up to pressures of 30 pounds per square inch above atmosphere.

The turbo-supercharger is a combination of a gas turbine and a centrifugal compressor, arranged as part of an airplane gasoline engine. The hot products of combustion from the engine exhaust are received upon the turbine runner and furnish power whereby is driven a centrifugal compressor mounted on the same shaft, which compresses air for supply to the carburetors. A more detailed description is given later.

In the latter part of 1917 the National Advisory Committee for Aeronautics requested the co-operation of the General Electric Company in the development of the turbo-supercharger in the United States. Our experience with gas turbines and centrifugal compressors led us to be greatly interested and the work was pushed vigorously during the war. An apparatus was constructed and placed in operation on an airplane engine near sea level. After a period of development, the stage was reached where nothing more could be done except at high altitude. However, since the development was not sufficiently advanced to warrant an airplane flight, the entire testing apparatus was taken to the summit of Pike's Peak. Here a further period of development took place. The apparatus was finally gotten into satisfactory working order so that the airplane engine developed the same power at the summit of Pike's Peak as it originally had near sea level. Arrangements had been started for installing the apparatus on an airplane when the Armistice intervened. Examination of the results which had been obtained, by Army officials after the Armistice, led to a resumption of the work and the apparatus was finally installed on an airplane. A very good showing was made from the first. The increase of power at high altitude was such as to give an entirely new set of conditions from those under which the airplane originally operated. This required various changes in the entire airplane apparatus and development was made of proper radiators, propellers, gasoline systems, cooling systems, etc. This work has been proceeding satisfactorily for some time.

Development work on the turbo-supercharger is also being carried on in France independently of our work. So far as can be seen from the published accounts of the French work, our apparatus is on a larger scale.

We are supercharging a larger airplane motor and are carrying the supercharging to higher altitudes. The mechanical details of the French and General Electric apparatus are quite different. The development of a turbo-supercharger similar to that of the French was started in this country but the design was modified considerably. Work on this apparatus has not been carried to a conclusion, however.

Our work was originally started at the suggestion of Dr. W. F. Durand, then Chairman of the National Advisory Committee for Aeronautics, who knew of our long experience with gas turbines and centrifugal compressors. It has since been carried on under the supervision at various times of Col. J. G. Vincent, Col. T. H. Bane, Major H. C. Marmon, Major G. E. A Hallett and Major R. W. Schroeder. Major Hallett has had charge of the development since the Armistice, and he has given considerable study to the matter of superchargers in general.

THE TURBO-SUPERCHARGER CYCLE.

Fig. 3 of the following article by Major Hallett gives a detailed diagram of the principles of the turbo-supercharger. The exhaust of the airplane engine is received by an exhaust manifold which leads it to a nozzle chamber carrying nozzles which discharge it onto the buckets of a turbine wheel. On the same shaft as this turbine wheel is the impeller of a centrifugal compressor. This compresses air from the low pressure atmosphere to approximately normal sea-level pressure and delivers it to an air discharge conduit which supplies the carburetors.

The turbine nozzles are of such area as to maintain within the exhaust manifold and nozzle box a pressure approximately equal to that at sea level. The difference between this pressure and the altitude low pressure gives a pressure drop for the exhaust gases which furnishes the power that operates the system.

Due to the respective temperatures, this power input suffices to give the desired compression and also to supply the inevitable losses. However, in order to avoid back pressure on the engine, above the normal sea-level value, both turbine and compressor must be designed with utmost attention to efficiency.

With an efficient arrangement, the engine when at high altitude exhausts at normal sea-level pressure and receives its air at the carburetor at normal sea-level pressure. Hence, normal sea-level power is delivered at all altitudes up to the maximum for which the supercharger is designed, so that the plane speed will increase uniformly as the altitude density decreases.

In order to reach this ideal there are various auxiliary problems that have to be solved; such as temperature rise of compression, slight deficiency of oxygen at high altitudes, effect of propeller on engine speed, and various other effects. The work thus far accomplished has demonstrated the validity of the fundamental principles and has disclosed the problems of detail.

MECHANICAL PROBLEMS OF SUPERCHARGING.

The General Electric superchargers thus far constructed have been designed to give sea-level absolute pressure at an altitude of 18,000 feet, which involves a compressor that doubles the absolute pressure of the air. This pressure ratio, with the quantity of air involved, requires about 50 shaft horsepower input for the compressor. The design of a complete power plant of this size to suit an existing airplane engine, with such weight and location as will not impair the flying characteristics of the

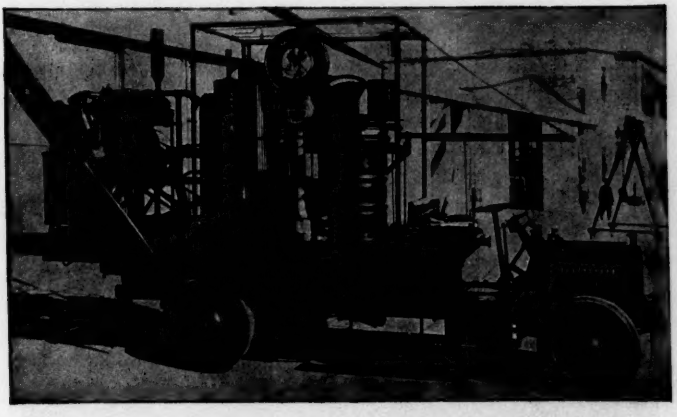


FIG. 1.—MOTOR TRUCK CARRYING LIBERTY MOTOR AND COMPLETE EQUIPMENT FOR TESTING THE GENERAL ELECTRIC SUPERCHARGER IN RARIFIED ATMOSPHERE AT PIKE'S PEAK.

FIG. 2.—PIKE'S PEAK SUMMIT ON WHICH EARLY TESTS WERE MADE OF THE GENERAL ELECTRIC SUPERCHARGER AT 14,109 FEET ALTITUDE.

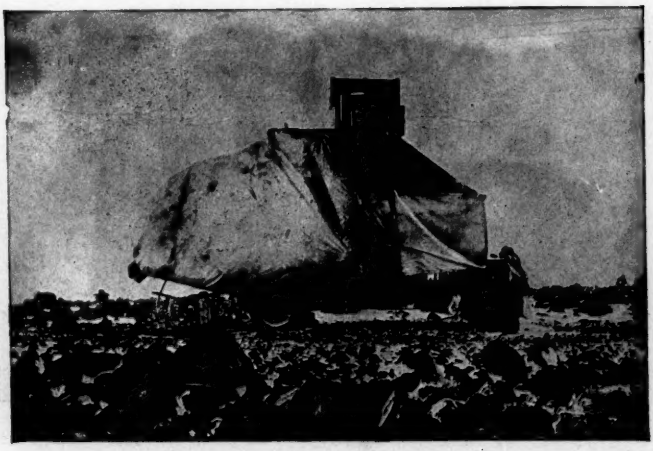


FIG. 3.—THE TEST CAR SHOWN IN FIG. 1, AS LEFT FOR THE NIGHT ON PIKE'S PEAK.

FIG. 4.—THE TEST CAR AS FOUND ON MANY OF THE MORNINGS.

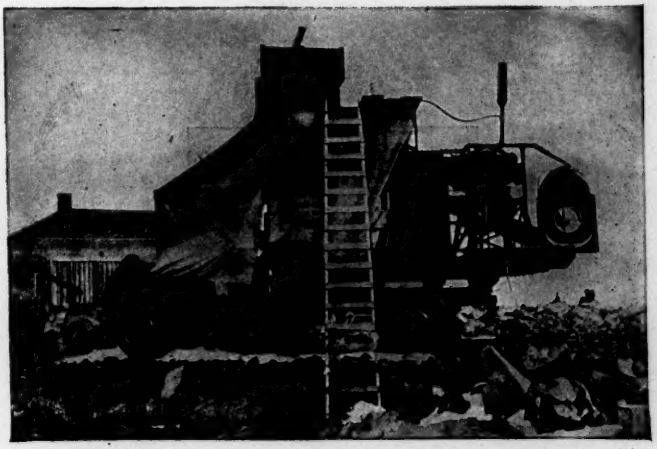


FIG. 5.—THE TEST CAR ON A PLEASANT DAY ON PIKE'S PEAK.

FIG. 6.—MAJOR R. W. SCHROEDER, LIEUT. G. W. ELSEY, AND THE AUTHOR (RIGHT TO LEFT.).

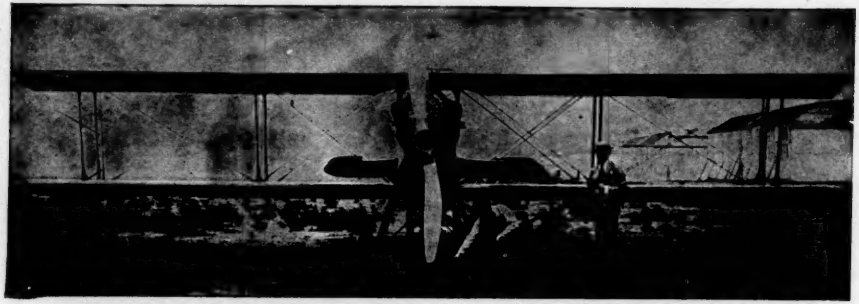


FIG. 7.—SUPERCHARGER EQUIPPED AIRPLANE. THE EXTRA LONG PROPELLER IS USED TO HOLD THE ENGINE SPEED DOWN TO NORMAL IN THE RARIFIED ATMOSPHERE OF HIGH ALTITUDE.

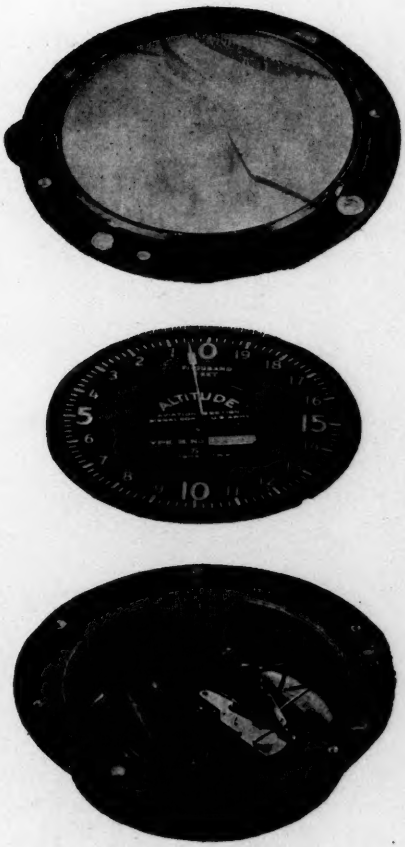
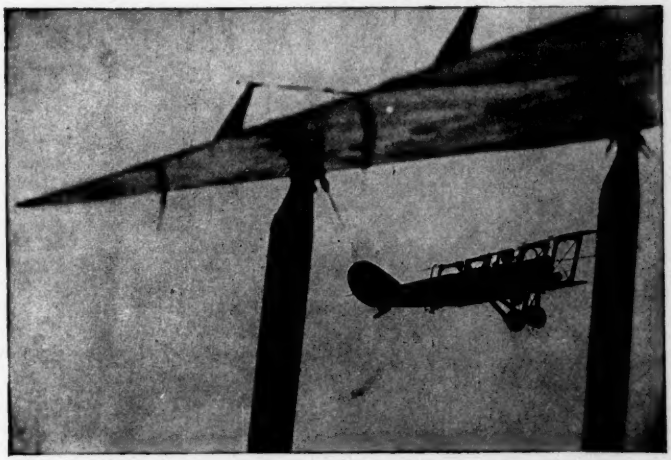


FIG. 8.—THE INSTRUMENT WHICH INDICATES TO THE AVIATOR THE HEIGHT AT WHICH HE IS FLYING.



General Electric Supercharger Equipped Le Pere Biplane Photographed from Another Plane While Test Flight at McCook Field, Dayton, Ohio.

plane, has of course offered many problems. The possibility of driving the compressor of the supercharger by engine power, instead of by the exhaust gases, suggested itself. Indeed, observation of an engine exhausting in the usual way into the atmosphere, discharging flames through the short red-hot spouts, with the almost incandescent exhaust valves in view, makes it seem absurd to propose to pass these red-hot gases through pipes with a pressure difference above the surrounding atmosphere equal to its absolute pressure, and more absurd still to obtain power by discharging these red-hot gases onto a turbine wheel rotating at 20,000 revolutions per minute. Nevertheless the turbo-supercharger has made flight after flight with entirely successful operation, while the mechanically-driven supercharger has never endured in spite of much effort. Much experience with the operation of the gas turbine led the writer to prefer its problems to those of the driving mechanism of supercharger operated from the engine. The turbine involves merely the addition to the compressor of a single extra wheel, designed for the conditions, with no extra bearings The engine-driven scheme involves a 50-horsepower transmission with multiplicity of gears, bearings, clutches, belts, and the like. These offer more or less drag on the engine when the supercharger is not in use at low altitudes, and very serious problems of acceleration when the supercharger is to be thrown into action, since the engine will be then running at its full speed of about 1,800 revolutions per minute.

It must be admitted that this is much the simpler proposition, since a turbine wheel has been designed which will endure.

The exhaust manifold and nozzle box have proven to be a very efficient exhaust muffler and conductor. Such a muffler and conductor is needed in any event, and the design of means for withstanding the increased pressure difference of the turbo-supercharger has been successfully accomplished.

POWER FOR TURBO AND ENGINE-DRIVEN SUPERCHARGERS.

An efficient turbo-supercharger theoretically deducts from the indicated horsepower of the airplane engine an amount corresponding to the difference between sea-level absolute pressure and altitude pressure. There is this additional back pressure during the exhaust stroke. The theoretical power available for driving the turbo-supercharger is greater than this, however, owing to the fact that there is available not only the energy due to the direct pressure difference mentioned, but also the energy of perfect expansion from the higher to the lower pressure. If there were no turbo-supercharger the engine would waste this energy in sudden pressure drop as the exhaust valves open. The turbine can utilize this energy. The sum of these two amounts of available energy, multiplied by the efficiency of the turbine wheel, gives the shaft power delivered to the compressor.

For an engine-driven supercharger compressor there is greater engine indicated power due to a lower exhaust pressure. However, the shaft power for the supercharger compressor must be transmitted through the engine connecting rod and crank shaft with losses, and then through the supercharger driving mechanism with additional losses. The total shaft power thus subtracted from the engine, multiplied by the efficiencies of these two transmissions, gives the shaft power delivered to the compressor. This is the same as for the turbo-supercharger. For a Liberty motor of about 400 horsepower and sea-level power at 16,000 feet altitude, this power is 50 horsepower.

The comparison then is as follows: The turbo-supercharger subtracts from the engine indicated power, adds power of expansion which would

not otherwise be used, and has turbine wheel losses. The engine-driven supercharger puts this indicated power through the engine (with some additional loads on the pins and bearings) and has engine and transmission losses.

With usual efficiency there is probably not a great difference between the gross subtraction from engine power in the two cases. There is then the disadvantage of transmitting the supercharger power through the engine pins and bearings, as well as through some mechanism between engine and supercharger, to be compared with the collection of the hot gases under pressure (with muffling advantages) and delivery to the turbine wheel. As already mentioned, practical success to date is in favor of the turbo-supercharger and the writer feels that this is really due to its innate superiority.

Engine-driven superchargers with positive-pressure blowers have been proposed. These have the additional disadvantage that with the desirable pressure ratios of about two to one there is an appreciable compression loss due to the fact that the machine only displaces air and has no direct means for compression.

It is to be noted that, although the power required to drive the supercharger is subtracted from the engine power, the remainder at high altitude with an efficient supercharger is equal to sea-level power. That is to say, the supercharged engine delivers power enough to drive the supercharger as well as to deliver sea-level power to the propeller. There is of course no way to arrange for full power due to supercharging with the additional power due to exhaust at the low absolute pressure of high altitude and without expenditure of power for supercharging. Without a supercharger the engine has the advantage of a very low exhaust pressure, but the explosive charge is so small that the gross power has the well known low value at high altitude.

SUPERCHARGING ENGINES.

Supercharging engines of various kinds, in which the engine crank case or the engine cylinders themselves are arranged for additional compression have been discussed by Major Hallett, and shown to give excessive weight and complication as compared with a turbo-supercharger.

A very simple form of supercharging has frequently been used wherein an engine of large displacement, but with very high compression pistons has been fitted to a comparatively small plane. In such a case, the throttle could not be opened wide near sea level because the compression would be excessive and serious pre-ignition would result; to say nothing of the damaging effect on the engine by delivery of the full power corresponding to the displacement with sea-level charge. At altitude, however, a full charge at the altitude density is taken, and on account of the high compression pistons this is compressed to a proper amount for good operation. Some high altitude flights have been made in this way with a single seat plane and engine with a displacement corresponding to 400-horsepower at sea level. The power at high altitude was possibly 100-horsepower. A 100-horsepower engine with a turbo-supercharger would give the same power at altitude and weigh very much less.

Since such an engine has normal compression pressure at high altitude, the power will be very nearly proportional to the density of the charge. There will be no loss of efficiency due to decrease of compression pressure. The altitude power will then vary directly with the cylinder displacement and inversely with relative density at altitude.

Major Hallett points out that with such an engine the weight is nearly proportional to the displacement. Hence such an engine will weigh nearly twice as much as a supercharged engine for 18,000 feet altitude conditions, and nearly four times as much for 35,000 feet conditions where the density is one-fourth that at sea level. There is some deduction from these figures due to the fact that the weight will not go up quite as fast as the displacement and because the supercharger's weight is not negligible. However, the situation in the main is as represented.

Engines have also been proposed with crank-case compression, either with individual connections or with a receiver. With a four-stroke cycle, two crank-case ends supercharge a single cylinder. However, with the minimum crank-case clearance thus far suggested, the maximum compression pressure possible is not sufficient to give supercharging at an appreciable altitude.

DESIGN OF GENERAL ELECTRIC SUPERCHARGERS.

The machines used thus far have been designed to give sea-level pressure at 18,000 feet altitude, which corresponds to a pressure ratio of about two. The rated speed for these conditions is 20,000 revolutions per minute. Sea-level pressure has readily been obtained up to 22,000 feet altitude. The control is entirely by hand operation of waste gates, which permits of free escape of some of the exhaust gases.

The entire apparatus, exclusive of exhaust manifold and air discharge conduit, weighs about 100 pounds. The exhaust manifold and air conduits have nearly the same weight as equivalent parts with no supercharger.

The turbine and compressor wheel have diameters somewhat less than a foot. The present design has been hampered by necessity for accommodation to existing engines and planes. It is proposed, however, to construct apparatus in which engine and supercharger are integral, with all parts arranged for the full possibilities of the combination.

The essential features of the design are various arrangements of ducts for cooling the several parts, means for accommodating the temperature expansions, means for handling the temperatures which exist, and design of both turbine and compressor to give utmost efficiency.

HISTORY OF GENERAL ELECTRIC SUPERCHARGER.

The combination of airplane, propeller, engine, radiator, cooling system, and supercharger are so intimately associated that no adequate tests can be made without the complete system in operation at full speed at altitude. Altitude chambers exist for tests of engines alone, but none are arranged for inclusion of the propeller. What tests were possible were first run with steam with the supercharger alone at the Lynn Works of the General Electric Company. Additional tests were run with the supercharger and Liberty motor on dynamometer stands at McCook Field, Dayton, Ohio, the Experimental Station of the Engineering Division of the Air Service. These tests were necessarily made with nearly sea-level initial pressure. Even slight supercharging under such conditions involved increase of compression pressure, and this instantly caused preignition. Both sets of tests gave means for perfecting the mechanical operation of the supercharger, but gave no information as to increase of engine power under altitude conditions. So far as could be seen everything was operating in accordance with scheduled expectations, but there was not sufficient assurance to warrant an airplane flight.

During the initial development of the Liberty motor a testing expedition had been sent to the summit of Pike's Peak, and it was decided to repeat this performance with the supercharger. Fig. 1 shows the motor truck that was prepared for the expedition. The Liberty motor carrying the supercharger was mounted on a cradle dynamometer, with scales and all arrangements for accurate measurement of power, gasoline consumption and the like. In fact, a complete testing laboratory was provided. The motor truck was shipped by rail to Colorado Springs, and then proceeded by its own power to Pike's Peak summit on the "Pike's Peak Auto Highway." This is a well constructed, very tortuous mountain road twenty-eight miles long.

Pike's Peak Summit has an altitude of 14,109 feet. It is the highest point in the United States easily reached by road. The summit is a slightly rounded rocky flat about 100 yards in diameter. On it are two stone houses, one at the terminus of a cog railroad and the other about one hundred yards distant at the terminus of the auto highway. The motor truck was set up near the latter. Fig. 2 shows the nature of Pike's Peak summit. Fig. 3 shows the way the test car was left after each day's work. Fig. 4 shows its condition on many of the mornings. There were, however, many pleasant days when the testing work could be carried on with facility. Fig. 5 shows the rear of the test car on a pleasant day.

The testing work at the summit lasted through September and half of October, 1918. The usual difficulties with experimental work were, of course, encountered with the addition of many delays, due to the cold and snow, and distance from repair shops. Minor changes were made in a little shack at the summit, but all the machine work and changes of appreciable magnitude were made at Colorado Springs. The apparatus was finally arranged to give good mechanical operation, and a number of tests were run showing the performance of the engine with the supercharger opened up to the maximum limit possible. The supercharger was designed for operation at 18,000 feet with some margin. It was possible at the existing altitude of 14,000 feet not only to supercharge so as to give full sea-level power, but also to overcharge so as to cause the engine to pre-ignite.

It was agreed that results of the Pike's Peak tests warranted the immediate installation of the supercharger on an airplane, and arrangements for doing this were in progress when the Armistice caused a cessation of the work. After the Armistice, careful re-examination of the situation resulted in resumption of the work in the early part of 1919. Various rearrangements were made in view of the experience gained at Pike's Peak and the apparatus was finally installed on an airplane. After a number of tests on the ground, flight tests were made.

It soon developed that a very appreciable increase of power was easily obtained when the supercharger was opened up. The whole airplane installation was not properly arranged to take advantage of this power, however, and changes were necessary in the radiator, cooling system, propeller system, gasoline tank, and pump system, etc. Changes in these parts have been made from time to time, and this work is still in progress. As the work proceeds more and more power is developed by the engine. Changes have also been made in the supercharger itself.

Many remarkable flight tests have been made. In fact, during the early work a flight record of some kind was broken at every flight. Appreciable progress has already been made, but the full capacities of the apparatus have not yet been reached, and further improvements of performance are to be expected.

Fig. 7 shows the airplane installation, and Fig. 6 shows Major R. W. Schroeder, who has made all of the flight tests to date, together with

Lieut. George W. Elsey, who has made all of the flight observations to date. The aviators are of course clothed for the intense cold of high altitudes and carry the parachutes that are now regularly used by the U. S. Air Service in experimental work.

MEASUREMENT OF ALTITUDE.

The altitude of an airplane is measured by an altimeter such as is shown in Fig. 8. This is essentially an aneroid barometer. It comprises a chamber almost wholly exhausted of air, on one side of which is a flexible metal diaphragm. As the atmospheric pressure presses on this diaphragm to a greater or lesser extent, the diaphragm moves in or out. This motion actuates a train of mechanism, ending in a needle moving over a scale. The temperature of the instrument itself must, of course, have no effect on the readings. Temperature compensation is arranged for by leaving a certain amount of air in the vacuum chamber and also by use of metal, in one of the levers, which has an appreciable coefficient of expansion. This temperature compensation is never quite exact, however, and a slight correction to the indications must be made, to take account of the actual temperature of the parts of the instrument at the time of an observation.

The reading of the instrument with temperature compensation taken into account gives the absolute pressure at the altitude in question and it is from this absolute pressure that the altitude is computed. Knowing the absolute pressure at the field from which the flight is made, as given by the barometer, the absolute pressure at altitude as given by the altimeter reading, and the temperature of the column of air between these two points at a number of heights, the difference in elevation can be computed by appropriate formula. There exist tables of average values of temperatures at various altitudes to enable this computation to be made approximately for an average case. However, where an actual altitude record is involved, the actual temperatures at various altitudes during the ascent must be observed and inserted in the formula. The determination of the altitude in a record flight is therefore a matter of some complexity. It has been very carefully done in the case of the supercharger flights at McCook Field.

The instrument in Fig. 8 is an indicating instrument. The instruments actually used for the final computation of altitude records are recording instruments called "barographs," which operate on the same principle. Fig. 10 shows the autographic record of such an instrument. After a flight the recording instruments used are removed from the plane and placed under the bell-jar of an air pump, connected with a mercury column, while the clock which causes the rotation of the record paper is still running. Autographic records are thus obtained at a number of known values of absolute pressure, as shown by the mercury column. This gives an accurate calibration and establishes the absolute pressure at the maximum altitude attained. During record flights three independent barographs are used for certainty.

Fig. 9. shows observations of temperatures at high altitudes for a great many of the supercharger flights. From the actual values of these temperatures for a given flight and the barograph record mentioned, the maximum altitude is computed.

The amount of supercharging is measured by a recording barograph of the same kind, which is not exposed to atmospheric pressure, however, but is enclosed in a sealed chamber connected by a pipe line to the air conduit at the carburetor inlets. By means of the known temperatures

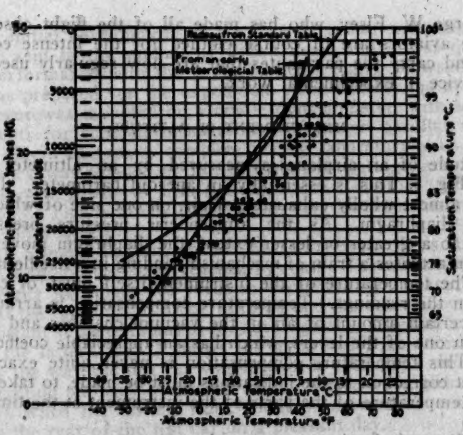


FIG. 9.—CURVES OF TEMPERATURE AT HIGH ALTITUDES. THE PLOTTED POINTS WERE MADE DURING SUPERCHARGER AIRPLANE FLIGHTS.

the altitudes corresponding to this record are known, so that there is given a record of the equivalent altitude of the engine. This is practically sea level as is shown by the lower curve in Fig. 10.

The upper curve in Fig. 10 gives readings of a Venturi-meter-Pitot-tube arrangement, which gives the air speed. These readings are calibrated by an actual flight near the ground over a measured course of three miles with the use of stop watches.

By these methods very accurate knowledge has been obtained of the performance of the supercharger under many conditions.

SUPERCHARGER PERFORMANCES.

The supercharger which has been used to date was primarily designed for high speeds at altitudes of 18,000 to 22,000 feet. The Le Pere plane on which the installation was made had a ceiling of about 20,000 feet with two men, and a speed at this altitude of 70 miles per hour. With the supercharger in use, a speed of about 140 miles an hour has been attained at 22,000 feet. As already pointed out, this has been attained with various parts of the plane installation in a partially developed state. Theoretical computations have been made showing that much higher speeds at high altitudes are to be expected. The progress of the flight tests to date indicates that the theoretical expectations will be fully realized.

The making of high altitude records has been very attractive and the supercharger has, of course, been used for this purpose as well as for the speed courses mentioned. Successively higher altitudes have been reached as experience has been gained regarding the manipulation of oxygen, gasoline, and other details. The highest altitude reached with two men was on October 4, 1919, with Major R. W. Schroeder and Lieut. George W. Elsey. The maximum indicated altitude was 32,335 feet. Various com-

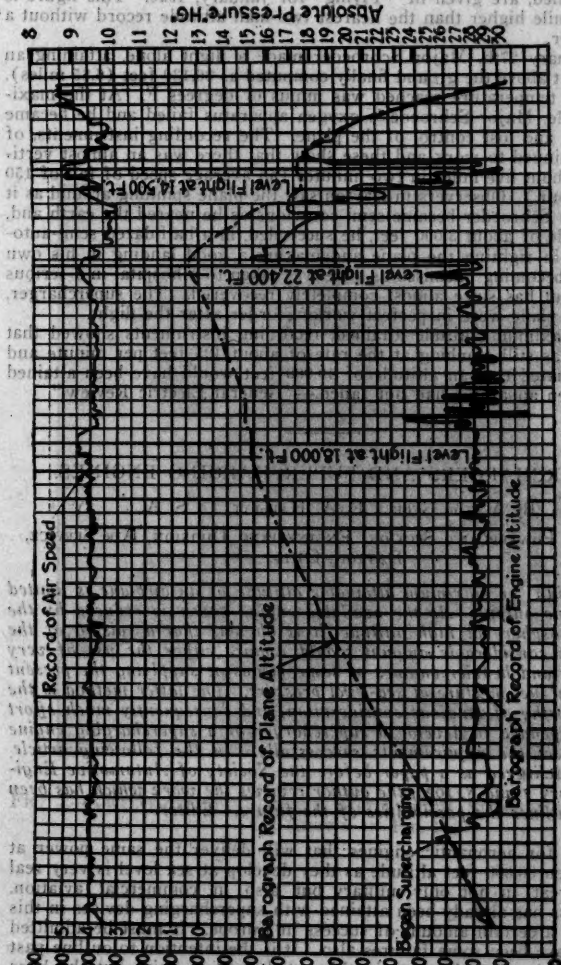


FIG. 10.—SAMPLE BAROGRAPH CURVE RECORD OF AN AIRPLANE FLIGHT. FROM SUCH A RECORD, AND THE TEMPERATURES AS SHOWN IN FIG. 9, THE TRUE ALTITUDES ARE CALCULATED.

putations from very complete observations, give the actual height above the ground as 31,800 feet. Complete details of these computations, as officially verified, are given in " Flying" for January, 1920. This figure is about one mile higher than the nearest two-man altitude record without a supercharger.

On February 27th, Major Schroeder made a flight alone, attaining an actual height above the ground finally computed as 36,130 feet (6.85 miles). The lowest temperature reached was minus 67 degrees F. At the maximum altitude, Major Schroeder's oxygen apparatus failed and he became unconscious and lost control of the plane. The recording instruments, of course, continued to work and these show that there was an almost vertical fall of about five miles in two minutes (an average speed of fall of 150 miles per hour.) Observers in Dayton saw the plane spinning around as it fell. Major Schroeder became semi-conscious as he neared the earth and, at an altitude of about 3,000 feet, he succeeded, in a half dazed semi-automatic way, in righting the plane and making a good landing in his own field, again becoming unconscious. He was taken to a hospital in a serious condition, but has since almost completely recovered. The supercharger, engine, and plane were in perfect working order after the flight.

At the maximum altitude attained, recording instruments showed that the plane was still climbing at the rate of about 125 feet per minute and it was estimated that an altitude of 40,000 feet would have been attained if the oxygen apparatus had not failed.—" General Electric Review."

SUPERCHARGERS AND SUPERCHARGING ENGINES.

By Major George E. A. Hallett, U. S. A.,

Chief of Powerplant Section, Engineering Division, Air Service, Dayton, Ohio.

The "ceiling" or maximum attainable altitude of an airplane is limited by the engine output. As the available engine power is reduced by the rarified atmosphere at high altitude, it is obvious that a raising of the ceiling by a considerable amount would require either the use of very much larger and heavier engines or some means of supplying the present engines with fuel mixture at sea-level pressure. The latter method is the only one worthy of studious consideration, and consequently much effort has been expended to develop a supercharger or a supercharging engine that will meet the requirements successfully. In the following article, which was delivered as a paper before the Society of Automotive Engineers, January 7 and 8, 1920, the author reviews the work which has been done and outlines the possibilities of the future.—Editor.

The need for aeronautic engines that will deliver the same power at 20,000 or even 30,000 feet altitude as they develop at sea level is very real and very great, in not only military but also in commercial aviation. Much success has already been attained with supercharging devices in this country and a certain amount of success in Europe. It must be admitted that there have been some failures also. It is the intention to outline past developments in supercharging in this article and to point out the lines of attack which seem to be meeting with most success.

Supercharging, as the term is generally used, means forcing a charge of greater volume than that which is normally drawn into the cylinders by the suction of the pistons in conventional internal-combustion engines.

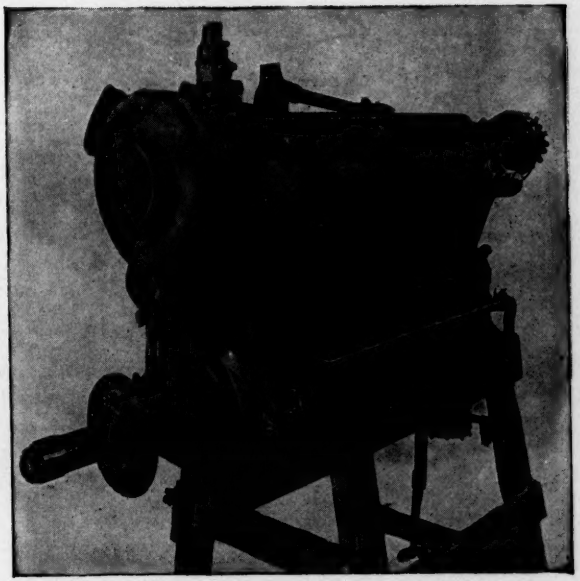


FIG. 1.—TWELVE-CYLINDER LIBERTY MOTOR WITH SUPERCHARGER OF THE TYPE SHOWN IN FIG. 3.

WHEN SUPERCHARGING IS NEEDED.

At 20,000 feet altitude the atmospheric pressure is roughly one-half that at sea level; hence about one-half the weight of charge is drawn into the engine and less than one-half the power is developed. At 25,000 feet altitude less than 25 per cent of sea-level power is delivered. If at these altitudes air is supplied to the carburetor at sea-level pressure, or approximately 14.7 pounds per square inch absolute, the power developed by the engine becomes approximately the same as when running at sea level. The low atmospheric pressure and density at great altitudes offer greatly reduced resistance to high airplane speeds; hence the same power that will drive a plane at a speed of 120 miles per hour at sea level will drive it much faster at 20,000 feet, and still faster at 30,000 feet altitude, *and with approximately the same consumption of fuel per horsepower hour.*

There is little to be gained by supercharging at sea level to increase the power of a given size engine, because the clearance volume must be made greater than normal to prevent pre-ignition, with consequent decrease in the expansion ratio and comparatively poor fuel economy. The fact that the clearance volume is increased removes the possibility of the engine developing full power at great altitudes unless a supercharging capacity greater than anything heretofore considered feasible is available. Supercharging, therefore, is most useful in maintaining sea-level horsepower in engines ascending to or working at great altitudes.

SUPERCHARGERS.

Superchargers usually take the form of a mechanical blower or pump and, of course, require a driving gear of some kind. The types of blowers or compressors used to date include the reciprocating, Root displacement, and centrifugal types. The reciprocating type was tried by the Royal Aircraft Factory early in the war, on an air-cooled R. A. F. engine, with practically no success. It seems that this type of blower was found to be comparatively heavy and also unsuitable, due to the pulsating pressure of the air delivered.

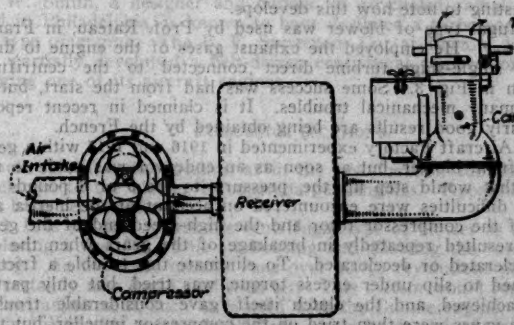


FIG. 2.—ROOT TYPE BLOWER SO ARRANGED THAT ITS AIR PULSATIONS SYNCHRONIZE WITH THE SUCTION STROKES OF THE ENGINE PISTONS.

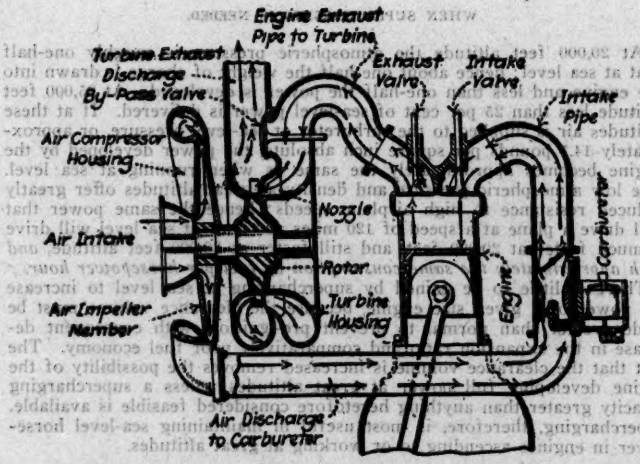


FIG. 3.—CENTRIFUGAL TYPE BLOWER DIRECT CONNECTED TO A TURBINE WHEEL THAT IS IMPELLED BY THE ENGINE EXHAUST GASES.

The Root type blower was tried by the Royal Aircraft Factory with little or no success. The trouble reported was "rough" running of the engine on account of the pressure pulsations in the air discharged by the blower, which tended to overcharge some cylinders and undercharge others, thus causing uneven impulses. It is reported that mechanical troubles also developed with this type of blower. George W. Lewis, of the National Advisory Committee for Aeronautics, is working on an improved Root type blower, shown in Fig. 2. Here the pulsations in the air discharged are synchronized with the suction strokes of the engine. It will be interesting to note how this develops.

The centrifugal type of blower was used by Prof. Rateau, in France, early in the war. He employed the exhaust gases of the engine to drive a high-speed single-stage turbine direct connected to the centrifugal blower shown in Fig. 3. Some success was had from the start, but he encountered many mechanical troubles. It is claimed in recent reports that some fairly good results are being obtained by the French.

The Royal Aircraft Factory experimented in 1916 and 1917 with a gear-driven centrifugal blower, but as soon as an endeavor was made to run it at speeds that would step up the pressure to the 5 or 6 pounds required, great difficulties were encountered on account of the inertia and momentum of the compressor rotor and the high-speed end of the gear-train, which resulted repeatedly in breakage of the gears when the engine was accelerated or decelerated. To eliminate this trouble a friction clutch, designed to slip under excess torque, was tried, but only partial success was achieved, and the clutch itself gave considerable trouble. Light, flexible vanes were then tried on the compressor impeller, but this expedient has not proved successful to date. Similar experiments were conducted by the A. E. F. in France, but were concluded by the signing of the armistice.

The United States Air Service started work on the Rateau type of turbo-compressor soon after we entered the war. The work was done under the supervision of E. H. Sherbondy, who worked in conjunction with the Rateau-Bateau-Smoot Co. which handled the Rateau patents in this country, and designed a turbo-compressor which seemingly embodied many improvements over the Rateau type. Three of these machines were built and given ground tests on Liberty engines. The arrangement of the engine and the supercharger is shown in Fig. 1. Considerable trouble was encountered due to overheating of the exhaust-driven turbine, and even the use of a special heat-resisting metal in this part did not overcome the trouble. Soon after Mr. Sherbondy began work on the turbo-compressor, Dr. S. A. Moss, of the turbine research department of the General Electric Company, was asked to carry on some work on the same general type. He built one turbo-compressor which was also a modification of the Rateau type, but differed considerably from Mr. Sherbondy's machine. This device was tested on a Liberty engine at the summit of Pike's Peak and developed approximately sea-level horsepower there, at an altitude of 14,000 feet. It was capable of making the engine pre-ignite at that height.

After the Armistice was signed all work on the development of superchargers was stopped. When the engineering division of the Air Service took over McCook Field and started to plan peace-time development, the supercharger situation was carefully considered. It was decided that it was important to continue development work along this line. It then became necessary to decide whether work should be continued on both the Sherbondy and the Moss machines, and, if not, which one should be developed. It was noted that although Dr. Moss' machine was comparatively crude, it contained some inherent advantages over the Sherbondy type, and no way was seen to overcome the faults of the Sherbondy machine. Therefore, the latter was dropped and the General Electric Company was given a contract to rebuild the old supercharger designed by Dr. Moss. The new device is now being tested in actual flight and is giving very interesting results. Figures on the results obtained with the present Moss supercharger are naturally confidential. The indications are that the turbo-compressor is very durable and probably will outlast an aviation engine.

J. W. Smith, a designer and builder of air-cooled radial engines, located in Philadelphia, is known to have designed a turbo-compressor for this type of engine. The B. F. Sturtevant Co., at Boston, Mass., has at least partially developed a belt-driven centrifugal compressor for supercharging one of its aircraft engines.

CARBURETOR LOCATIONS.

There is still some question as to the best location for the carburetor in relation to the blower in supercharged engines. Apparently all positions have been tried:

(1) It is possible to use the centrifugal type of blower as a carburetor by placing a fuel jet within its housing and allowing the rotor to do the mixing. As the rotor usually runs over 20,000 revolutions per minute, it will certainly mix liquid fuel with air. This system would require a manual fuel adjustment, such as is used with the Gnome engine, for different speeds. With this arrangement there would be danger of an explosion in the blower in case the engine back-fired, because the mixture in the blower would be under pressure higher than atmospheric.

(2) The carburetor can be placed on the suction side of the blower. In this case the evaporation of the fuel will assist in cooling the charge during compression and the action of the compressor will improve the mixing of the fuel, but the danger from explosion remains to be overcome.

(3) When the carburetor is placed in the "normal" position and air is forced through it, it becomes necessary to "balance" the float-chamber with supercharger pressure. This somewhat complicates the feeding of fuel. Pressure gas-feed systems are "banned" in military planes and in any case with a pressure system the tanks would have to be made comparatively heavy to withstand the pressure which would be used at great altitudes. Where gasoline pumps are used it is necessary to regulate their discharge pressure as the plane ascends, because the fuel must reach the float-chamber at a pressure about 2½ pounds higher than that at the supercharger outlet. If the difference in fuel and float-chamber pressures is not kept in constant relation, the quality of the mixture fed to the engine will vary on account of the change in fuel level in the float-chamber. The engineering division has developed a very simple device that solves this problem effectively and is entirely automatic.

It would naturally seem at first thought that the extremely low temperatures always found at great altitudes would make possible the easy solution of cooling problems, but in reality the low density of the air reduces its heat conductivity and capacity for heat absorption to such a point that a supercharged engine developing sea-level power at 20,000 feet requires a little *more cooling surface* than it does when developing normal power at sea level.

The Liberty engine and many others run best with a water temperature of about 170 degrees F. To maintain the cooling water at this temperature in the reduced atmospheric pressure at 25,000 feet it is necessary to use several pounds of air pressure in the radiator to prevent the water from boiling away. Very effective radiator shutters are needed when the engine is throttled to make a descent from altitudes of over 20,000 feet to prevent the water in the radiator from freezing before warmer air is reached.

Contrary to expectations, the Moss turbo-compressor now being tested at McCook Field does not complicate the pilot's controls. On a normal engine the pilot handles the throttle and the altitude carburetor control which thins down the mixture as he ascends. With the turbo-compressor the altitude control becomes unnecessary up to the altitude at which the engine can no longer deliver sea-level power but is used, as with a normal engine, if the plane is driven higher.

With the Moss turbo-compressor, when flying at low altitudes, the exhaust pressure is allowed to "waste" through manually operated "gates" in the exhaust pipes. As the plane ascends the pilot closes these gates a little at a time and after he reaches a great altitude he can speed and retard the plane by the use of these gates. He uses the throttle only in case he wants to descend rapidly, when he closes it. In our test flights we have provided the pilot with a sealed altimeter connected only to the supercharger pressure, so that it shows to what altitude this pressure corresponds. When at great altitude the pilot closes the exhaust gates until the pressure in the carburetors causes the altimeter to show sea-level pressure. This makes it unnecessary for him to do any calculating. If he makes the gauge read lower than sea level, the engine will pre-ignite. We have already been able to obtain sea-level pressure in the carburetors at well over 20,000 feet. The exact height cannot be mentioned at present.

With a normal engine the falling off in power as the plane ascends does not cause as much of a drop in propeller speed as might be expected, be-

cause of the reduction in density of the air in which the propeller is working. Our best engines do not lose over 75 revolutions per minute at 20,000 feet. When an engine is supercharged so that the power remains constant as the plane ascends, the propeller tends to "race" at great altitudes. Therefore it is necessary either to use a variable-pitch propeller or to put on one that holds the engine speed down too low for best performance near the ground, but also does not allow the engine to race too much at great altitude. In our present tests we are using an oversize propeller and are getting surprisingly good results, but we also have variable-pitch propellers about ready for test and should get much better performance with them.

SUPERCHARGING ENGINES.

As generally used, the term "supercharging engines" refers to internal-combustion engines in which compression in the crankcase or in the lower end of the cylinders is used to force an additional volume of air or mixture into the working cylinders *after completion of their normal suction stroke*. Early in the war the Army and the Navy each placed an order with the Kessler Motor Co., Detroit, Mich., for several experimental supercharging engines. This type of engine, shown in Fig. 4, supercharged

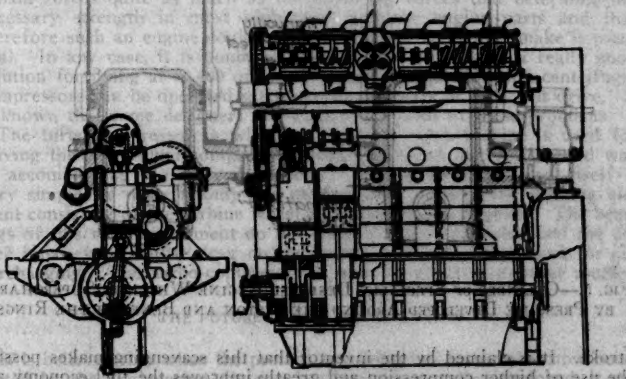


FIG. 4.—SPECIALLY DESIGNED ENGINE WHICH IS SUPERCHARGED BY PRESSURE DEVELOPED IN THE CRANKCASE.

each cylinder by the use of crankcase pressure, as is possible in four-cycle engines. Experiments were made using both air and mixture in the crankcase. Considerable difficulty was encountered in both the design and construction of the engine and so far as the engineering division has learned, no complete tests have been run; and in the small amount of testing that has been done no very large increase in power or brake mean effective pressure has been shown officially. It is believed that the frictional losses will prove to be very high in this type of engine and that the supercharging will be comparatively limited. A similar engine which was tested in this country did show very high frictional loss, due partly to the work of operating the valves which controlled the crankcase air.

An interesting problem in this type of engine when using air in the crankcase is whether a rich mixture should be fed through the regular induction system and an effort made to dilute it with the supercharged air, or a normal mixture should be fed through the induction system and an attempt made to obtain perfect stratification and thus let the supercharged air merely form a cool, elastic and expanding cushion on the piston-head. It is feared that in either case it will be difficult to secure the desired results through a large range of speeds and throttle positions.

There is an English make of supercharging engine in which air is compressed under the piston and by-pressed through cylinder ports at the bottom of every stroke (see Fig. 5), supercharging, as in the Kessler engine, at the end of the suction stroke and scavenging at the end of the exhaust

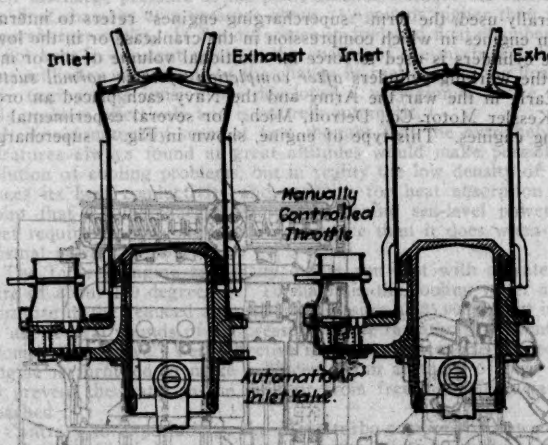


FIG. 5.—CYLINDER OF SPECIALLY DESIGNED ENGINE WHICH IS SUPERCHARGED BY PRESSURE DEVELOPED AROUND THE PISTON AND BENEATH THE RINGS.

stroke. It is claimed by the inventor that this scavenging makes possible the use of higher compression and greatly improves the fuel economy and brake mean effective pressure. It is believed that this engine will give rather limited supercharging and it may prove difficult to control the mixing or stratification of the air and mixture at some speeds.

In an English rotary air-cooled engine the pistons travel out to the cylinder-heads on the scavenging stroke and the beginning of the suction stroke and continue an extra distance inward at the end of the suction stroke, thus taking in a larger charge than that of a conventional engine. The piston reaches only a normal position at the end of the compression stroke and continues an extra distance inward at the end of the suction stroke, all by means of an eccentric crankpin bearing which is rotated on the crankpin by gears of suitable ratio. This type of engine must certainly give a very limited amount of supercharging.

It is believed that supercharging engines will necessarily give a rather limited amount of supercharging. It is also believed that considerable difficulty will be encountered in obtaining the desired stratification in

mixing conditions in the combustion chamber through any wide range of throttle positions. Also, some mechanical friction is added in this type of engine and it must be borne in mind that friction is particularly undesirable at great altitudes because it remains nearly constant from the ground up to great altitudes while the power falls off rapidly; therefore, the mechanical efficiency of the engine becomes very low.

The Root type of blower might be interesting for supercharging purposes if the troubles caused by the pulsating nature of its discharge could be eliminated. It is hoped that Mr. Lewis' efforts along this line will meet with success.

It is already frequent practice to build aviation engines with compression so high that the throttle cannot be fully opened on the ground without injury to the engine. In this way, perhaps, the same power is obtained at 5,000 feet as can be obtained on the ground. It has been suggested that this idea be carried further and that an "oversize" engine be built with much higher compression so that the throttle cannot be opened fully until a considerable altitude, such as 10,000 or 15,000 feet, is reached. It has been stated that such an engine could be made lighter, in proportion to the cylinder sizes, than a conventional engine, on account of the fact that the throttle would never be opened near the ground, but it is believed that when this idea is investigated, it will be found that it is the inertia forces quite as much as the explosion forces that determine the necessary strength in most high-speed airplane engine parts and that therefore such an engine could not be built light enough to make it practical. In any case, it is doubtful whether this would give a really good solution for flying at 25,000 or 30,000 feet. It is possible that centrifugal compressors can be operated satisfactorily by gears or by a belt drive. It is known that some designers are working on both of these problems.

The turbo-compressor in which an exhaust-driven turbine is used for driving the centrifugal compressor seems to present one fairly good way of accomplishing the desired purpose. The turbo-compressor itself is very simple, as there is only one moving part, namely, the rotating element consisting of the turbine wheel and compression impeller. The bearings of this rotating element do not seem to wear noticeably and the device imposes very little drag on the engine when not being used for supercharging. The turbo-compressor is also an effective exhaust muffler.

THE FUTURE OF THE SUPERCHARGER.

It is believed that when the present type of turbo-compressor now being tested by the engineering division has been more fully developed, it can be built into an engine in a form which will add less weight and less head-resistance than the present machine, and naturally when we know exactly what additional cooling surface is required at a given height, it will not be difficult to build this cooling surface into the airplane in such a form that very little weight and head-resistance will be added.

The uses of the supercharger for military service can be divided into; first, for airplanes in which it is desired to reach extreme altitude; second, for airplanes in which it is desired to increase the rate of climb and horizontal speed and therefore maneuverability at altitudes where it is intended to fight; and, third, for airplanes which carry large loads, such as bombers, which normally are handicapped by having a very low ceiling and whose entire usefulness would, if larger engines were installed to pull them to a higher ceiling, be lost on account of the larger amount of fuel and other material that would have to be carried, thus decreasing their radii of action.

In the first case it is believed that a special supercharger can be built that will make feasible much greater altitudes than any that have been attained with the present General Electric turbo-compressor; and it is considered essential that we have airplanes capable of reaching very great heights. In the second case, it is pointed out that military machines not fitted with supercharging engines, when fighting at an altitude of 20,000 feet or more, are so near their ceiling that their rates of climb, speed, and maneuverability are comparatively poor, but the use of a supercharger seems to overcome this difficulty easily. When a pilot climbs with a normal engine to 20,000 feet and then levels off in horizontal flight, the engine and propeller speed up perhaps 100 revolutions per minute. This, of course, enables the engine to develop slightly more power. In the case of a supercharged engine, especially with the turbo-compressor type of supercharger, as the engine speeds up in horizontal flight, the temperature of the exhaust and the power available from the exhaust increase, thus building up the supercharging pressure and giving considerably greater increased power than with a normal engine.

The use of superchargers in commercial airplanes of the future is assured because superchargers will make possible far *more miles per hour* and *more miles per gallon* with a given engine and airplane, and speed is the main advantage of air over other kinds of transportation. It is thought by many qualified judges that, by flying at a sufficient height with a supercharged engine and a suitably designed airplane, a speed of 200 miles per hour can be maintained.

In the heavy-load-carrying type of plane which must necessarily cross mountains or perhaps fly above storms and clouds, the necessary height can be reached with smaller, cheaper, and more economical engines if they are fitted with superchargers. It is obvious that in really long cross-country flights or trans-continental flights, with mail or passengers, the logical course is to fly at 25,000 or 30,000 feet altitude where the resistance to speed is low and great speed can therefore be attained provided the engine can deliver high power economically, which it can do if equipped with a supercharger.

As a graphic illustration of the advantage of a supercharged engine, it is pointed out that at 25,000 feet altitude a supercharged 250-horsepower engine will deliver as much power as a 1,000-horsepower engine without a supercharger; and of course the former will weigh many hundred pounds less, its fuel and tankage will weigh very much less, the first cost will be much lower and the structure of the airplane can be made much lighter.—"General Electric Review."

EROSION OF PROPELLERS.

It will be recalled that at last year's meeting of the Institution of Naval Architects a paper was presented by Sir Charles A. Parsons and Mr. Stanley S. Cook entitled "Investigations Into the Causes of Corrosion or Erosion of Propellers," an abstract of this paper being given in "Shipbuilding and Shipping Record" of April 17, 1919. The theory was advanced that rapid and deep erosion of propeller blades and the denting of bosses might be due to the water-hammer effect produced by the closing up of vortex cavities. That the blow produced by the rapid collapse of a vortex cavity is sufficient to produce a destructive effect on a metal was clearly demonstrated by Sir Charles Parsons at the conversazione of the Royal Society at Burlington House, when he exhibited the apparatus which was employed by the sub-committee of the Board of In-

vention and Research to investigate the subject of propeller erosion. A hollow cone is fitted at its apex, with a die cap through which is a hole of the same diameter as the end of the cone, a thin metal plate being placed between the cap and the cone. The cone is placed in water in a tall tank and after being filled with water it is thrust quickly downward on to a block of rubber at the bottom. This momentarily forms a cavity at the apex of the cone, which closes with a perceptible metallic click and with sufficient force to puncture the metal plate. The plate is 0.03 inch thick, piercing of which indicates a pressure of 140 tons per square inch. It is difficult to see how the effect of these collapsing vortices on propeller blades can be resisted.—"Shipbuilding and Shipping Record."

ECONOMICAL PROPELLING MACHINERY.

Some years ago an interesting discussion took place in the transactions of the N.E. Coast Institution of Engineers and Shipbuilders, as to the merits of reciprocating machinery, geared turbines and oil engines. It is evident that the difference of opinion then expressed still exists amongst shipowners, for if experience had shown that either one or the other had outstanding merit, it is certain that by this time the application of that particular type would have been more universal. Electric propulsion has come prominently before the engineering public since the above discussion took place, and thus four distinct types, with many variations, may be chosen from. It is possible there may be a distinct field for each. It is certain, for example, that vessels demanding great power for the desired speed must have either geared turbines or the electric drive. The advocates of each are equally strong in their claims for greater efficiency. On the one hand the electric drive has been criticised as "being far outclassed in every respect and for all sizes of ships by the geared turbines," whilst on the other "fuel economy decidedly superior to anything produced in a steamship" is claimed for electric machinery. It is difficult for the ordinary man to judge where the truth lies. When slow cargo ships are considered, and at least 60 per cent of the world's tonnage comes under the category of low-powered ships, the nature of the trade and the average route should be considered. If oil can be procured cheaply, there seems to be much in favor of the internal-combustion engine, in spite of its greater initial cost and greater call for skilled attention. Oil fuelled Scotch boilers with geared turbine machinery are being installed in many cargo ships at the present time. A large amount of valuable data as to relative merits of the various types should soon be at the disposal of engineers and shipbuilders.—"Shipbuilding and Shipping Record."

NAVAL OIL STATIONS.

Considerable sums are put down in the Naval Works estimates for oil fuel depots in various parts of the world. Money is to be spent on such depots at Gibraltar, Hong Kong, Jamaica, Malta, Plymouth, Portland, Port Said, Rosyth, Port Edgar and Glasgow. This enterprize marks the passing of the Navy more and more to a basis of oil-driven ships. Every new ship turned out today, says the First Lord of the Admiralty, is an oil-driven ship, and therefore stores of oil must be present at every Naval station which our Navy uses all round the world. We used to speak of coaling stations. Now we are beginning to speak of oil fuel stations.

At present, this country controls but a few of the oil-producing areas, and therefore the Admiralty take the view that we should have stores of oil accumulated at selected points in view of any emergency that may arise. There has been a tendency to criticize some of the points which have been selected as oil stations, but the First Lord puts forward the contention that if a depot is not provided, the Navy will have to have recourse to an oil hulk at sea, which is an expensive method of storage. The appearance of these items in the estimates, however, are significant of the official anticipations regarding the future of oil fuel.—"Shipbuilding and Shipping Record."

"BUILDING UP" BY WELDING.

There appears to be no limit to the number of successful applications of the principles of electric welding in marine engineering and ship construction. Apart from the specific purpose of welding two pieces of material together to form a continuous whole, the idea of depositing metal at a worn part from a suitable electrode has long been practiced. This process has been extensively employed in boiler work where the plates have corroded and in machine parts where it is required to add strength to a piece of material which shows signs of weakening due to wear. We recently had an opportunity of witnessing this process being applied to a large crankshaft, which, as a result of many years of wear, had become slack in its web. It was decided to build up the diameter of the shaft by welding a thin layer of metal all round for a length equal to the thickness of the web and afterwards to turn the shaft down to size and shrink the web on to the shaft. This was successfully accomplished, and although it must be admitted that the process of building up the shaft was a slow one, the welded metal turned down to the correct diameter giving the appearance of a new and homogeneous shaft. In this case the electrodes were operated by hand, but the use of machinery for this purpose would greatly reduce the time taken and would lessen also the amount of material which had to be used in building up the shaft, since the welded metal could be laid more uniformly on the shaft.—"Shipbuilding and Shipping Record."

BAKELITE PROPELLER TEST.

Major William Ocker, flying a DH-4 Goldbug from Aberdeen, Md., to Washington, D. C., equipped with the new Micarta or so-called Bakelite propeller, put this propeller to a most severe test. Bolling Field had been in a very bad condition for about two weeks, the mud being 12 inches deep. Landing with a DH-4, which has a very high landing speed, in mud is a particularly dangerous undertaking. Hardly had his wheels touched the ground when his plane nosed over. Before he could reach the switch to cut off his engine the propeller succeeded in churning around in the ground several times. The force of this churning was so great that it threw his plane back into position, while the wheels sank down to the hub in the mud. When an examination of the propeller was made it was found to be in perfect condition and not even chipped. If this had been a wooden propeller it would have been broken into match sticks.—"Scientific American."

BOOK REVIEWS.

DETERIORATION OF STRUCTURES IN SEA-WATER. FIRST REPORT OF THE COMMITTEE OF THE INSTITUTION OF CIVIL ENGINEERS. Edited by P. M. CROSTHWAITE, B.A.I., M.Inst.C.E., and GILBERT R. REDGRAVE, Assoc.Inst.C.E. London: The Committee of the Institution of Civil Engineers, Great George Street, Westminster, S.W.1., and to be purchased through any bookseller or directly from H.M. Stationery Office, Imperial House, Kingsway, W.C.2, and usual agencies. 6¼ in. × 10 in. × 1¼ in. 301 pp. Price 30s. net.

The deterioration of structures in sea water is a subject of vast importance to the waterfront engineer, as affecting repairs to existing structures and all future designs. Six million dollars is the estimated cost of repairs due to action of the teredo in the timber constructions at an important waterfront property which for 50 years was free from any sign of this destructive agency. Many failures, throughout the world, of concrete structures in sea water further emphasize the importance of the subject.

In July, 1916, the Institution of Civil Engineers of Great Britain took steps to secure a grant from the Privy Council in aid of research work into the deterioration of structures of timber, metal and concrete, when exposed to the action of sea water in various localities and as to the measures for preservation of such structures.

The present volume is the first fruit of the undertaking. The Institution of Civil Engineers has unusual opportunities

for conducting such investigations and early appointed some forty willing representatives in the persons of the Chief Engineers of Ports in the United Kingdom, India, the Dominion, and Colonies. All parts of the world and varying climatic conditions are thus represented.

An excellent resumé of existing literature on the subject, principally from British sources, is given, but touching also on American, Dutch, German and other experiences and investigations. Special reports from the corresponding members are printed, covering British, African, Australian, New Zealand, East Indian, Chinese (British) and North American (Canadian) ports.

Particularly complete and valuable are the summaries and discussions of the wood boring organisms, the selection of timbers to resist their action, the behavior of concrete in sea water, the causes of its deterioration, and the action of cast iron, wrought iron and steel in sea water.

The reports from the various ports are illustrated with plans and photographs, and are individually well worth careful study by those charged with responsibility for waterfront structures. To a very great extent the British experiences are paralleled by those of American engineers.

The present report by no means exhausts the subject and the Committee of the Institution proposes to proceed on the lines which the present information indicates have not been fully covered. The work of the coming generation of engineers, as well as of the present engineers for waterfront constructions, will depend to an unusually great extent upon the world-wide experiences of engineers in actual constructions, but unless these experiences are comprehensively collated and analyzed the engineering profession will not get the full benefit. The undertaking of the Institution of Civil Engineers is thus particularly deserving of commendation.

R. E. B.

HOT BULB OIL ENGINES AND SUITABLE VESSELS. By WALTER POLLOCK, M. I. N. A., M.I.Mar.E., M.I.Mech.E., M.I.M. D. VAN NOSTRAND COMPANY, 8 Warren St., New York. 429 pp., 369 ill. $10.00 net.

For purposes of marine propulsion in powers up to 300 horsepower per engine and for stationary purposes, such as driving electric generators, etc., up to 150 horsepower per engine, the semi-Diesel or hot bulb engine is sometimes considered as being superior to the Diesel engine, but in powers in excess of those given above, it is considered that the Diesel engine is superior in every way.

The field designated above is a large one and this, together with the greatly increased numbers of engines of the hot bulb type in use, merits special attention being given to this subject and this volume supplies a long-felt want.

The book is most valuable from the point of view of the owner and the operator rather than from that of the designer. The text goes briefly into principles, thoroughly into descriptions of the various makes of this type of engine, and, most important of all, is the information which it gives to a prospective user in choosing an engine suitable for his particular purposes.

The notes on upkeep and operation are good and indicate, on the part of the author, a wider acquaintance with the semi-Diesel than is afforded in the drafting room, and as there is a great lack of published information on these subjects, the present volume is a welcome and needed one.

M. C. B.

DIESEL ENGINE DESIGN. By H. F. P. PURDAY, B.Sc., A.C.G.I. D. VAN NOSTRAND COMPANY, 8 Warren St., New York. 295 pp. 271 ill. $7.50 net.

With the renewed interest in the Diesel engine for propulsion that has arisen to meet the requirements of cheaper modes of transportation, is the demand for suitable technical works on the design and operation of this type of prime mover.

The volume in question is very complete from the point of view of the drafting room and presents an exhaustive number of examples of good practice in every detail of the design of a Diesel engine. The chapters on first principles and theory are especially good and are presented in simple language. The numerous illustrations are clear and well chosen, the tables are excellent, and the references at the end of each chapter are exceedingly helpful.

This is an excellent book which can be used with confidence by designers and draftsmen in Diesel engine work.

M. C. B.

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The following members and associates have joined the Society since the publication of the last JOURNAL:

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